356
6/19/79

DR. 2772

SAN-1592-1

COST-EFFECTIVE CONTROL SYSTEMS FOR SOLAR HEATING AND COOLING APPLICATIONS

Final Report

By
Jane H. Pejsa
William W. Bassett
Stephanie A. Wenzler
Khanh H. Nguyen
Thomas J. Olson

September 1978

MASTER

Work Performed Under Contract No. EG-77-C-03-1592

Honeywell, Inc.
Energy Resources Center
Minneapolis, Minnesota

U.S. Department of Energy

Solar Energy

DISTRIBUTION OF THIS DOCUMENT IS UNLIMITED

NOTICE

Available from the National Technical Information Service, U. S. Department of Commerce, Springfield, Virginia 22161.

Price: Paper Copy $9.00
Microfiche $3.00

SAN-1592-1
Distribution Category UC-59c

COST-EFFECTIVE CONTROL SYSTEMS FOR SOLAR HEATING AND COOLING APPLICATIONS

FINAL REPORT
September 1978

Jane H. Pejsa
William W. Bassett
Stephanie A. Wenzler
Khanh H. Nguyen
Thomas J. Olson

This work has been supported by the Solar Heating and Cooling Research and Development Branch, Office of Conservation and Solar Applications, U.S. Department of Energy.

Contract Number EG-77-C-03-1592

Honeywell
Energy Resources Center
2600 RIDGWAY PARKWAY,
MINNEAPOLIS, MINNESOTA 55413

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

PREFACE

As part of its research and development activities, the Department of Energy is supporting studies to develop control strategies and control subsystems that will lead to improved performance and cost effectiveness of solar heating and cooling systems.

The Honeywell Energy Resources Center is a participant in this effort. This final report summarizes a twelve-month study that has included an extensive literature and marketplace search as well as computer analyses involving six different solar heating and cooling system designs. Recommendations reported herein are for cost-effective control subsystems and improved system performance.

TABLE OF CONTENTS

TABLE OF CONTENTS (CONTINUED)

TABLE OF CONTENTS (CONCLUDED)

LIST OF FIGURES

LIST OF FIGURES (CONCLUDED)

LIST OF TABLES

LIST OF TABLES (CONCLUDED)

INTRODUCTION

Approximately one-fourth of this nation's energy use is directly associated with climate control in residential and commercial buildings.

In the past, when energy costs were not a significant factor, the level of system performance was not the foremost consideration in heating and cooling matters. Systems were designed to be either off or on and were often oversized, since operating efficiencies were of little concern. The level of sophistication of the associated controls was determined accordingly.

As the availability and costs of conventional energy become crucial, the level of system performance becomes increasingly critical.

On one side more sophisticated strategies and design criteria are being developed to conserve energy in new and existing conventional systems. On the other side solar collector systems are being incorporated into new heating and cooling applications.

Each of these developments has led to a rapid increase in the functions addressed by a control system and hence in its complexity and cost.

Concurrently with these new directions in control functions and complexity, developments in microprocessor--and more recently, microcomputer--technology are revolutionizing the control marketplace.

It is important to take a critical look at the effect of these multiple developments on the practicability of cost-effective control for solar-assisted heating and cooling systems. During the study documented herein, a methodology has been defined to arrive at control recommendations for a variety of climate control system designs, applications and regions, and the results are presented in two parts.

Part I consists of a literature and marketplace survey, involving control strategies, functions, sensors, actuators and the controllers themselves.

Part II represents the bulk of the study effort--an attempt to simulate and evaluate system performance for several representative residential and commercial heating and cooling system designs and thus to derive improved performance techniques within cost-effective control systems.

PART I. CONTROL SURVEY

A survey of the literature and marketplace has been completed to ascertain the state of the art, as well as recent advances in the design of controllers, addition of functions and the associated strategy innovations to maximize performance of heating and cooling systems using solar energy. This survey includes a compilation of cost and performance data to aid in the selection of strategies and functions evaluated under Part II of this study.

The survey results are reported under three categories:

- Control Strategies and Functions,
- Control Implementation,
- Control Mechanization.

Following these are some general conclusions and a complete list of references.

A. CONTROL STRATEGIES AND FUNCTIONS

A survey of the literature as well as of ongoing development work at Honeywell yields a number of control strategies and operating modes that are applicable to solar climate control systems.

Some of these strategies and operating modes are specific to the design of the solar heating and cooling systems, while others are more or less hardware independent. The following groups of strategies and modes appear to be significant enough to warrant documentation, evaluation and possible incorporation into the design of solar climate control systems:

- Proportional control:
  - Collector loop,
  - Storage loop,
  - Ambient air.
- Direct heating mode.
- Heat pump options.
- Thermostat setback and setup strategies:
  - Heating season,
  - Cooling season.
- Temperature reset strategies:
  - Collector loop,
  - Storage tank.

- Off-peak operating strategies:

  - Weather anticipation,
  - Storage management.

- Peak load strategies:

  - Load shedding,
  - Duty cycling,
  - Heat pump strategies.

1. Proportional Control

As a starting point it seems appropriate to isolate a single control tradeoff, the ramifications of which are exceedingly important to system design and control implementation. The smoldering issue of fixed on/off* control versus proportional control in the collector loop certainly qualifies as such a tradeoff. This issue is reflected in the various hardware packages on the market designed to harness solar energy and it is frequently addressed in papers relating to control optimization of solar heating and cooling systems. Therefore, the issue must be examined in both of these contexts: the control strategies and the hardware implementation.

There is precedent for either an on/off or a proportional control approach in the literature available.

---

* Also known as "slam-bang" or "bang-bang" control in various technical circles.

In work under way at the Los Alamos Scientific Laboratory, advanced control techniques such as adaptive control are being applied to the design of proportional control systems. Published material coming out of Los Alamos to date reveals on/off collector control within the adaptive control algorithms (References 1, 2 and 3).

> "The cost of running these pumps is small, so there is little to be gained in energy and much to be lost in complexity by using proportional control." (Reference 1).

Similarly, results of analysis work undertaken in Australia indicate that in systems with nonstratified storage

> "... if the amount of energy collected is the only criterion of performance, maximum flow rate is to be applied whenever there is a positive gain from the collector." (Reference 4).

On the other hand Rho Sigma, one of the leading manufacturers of solar system hardware and control components, has come to a contrary conclusion on this issue. (Reference 5). At Rho Sigma, collector performance was simulated under two control strategies:

(1) A fixed differential control limit ($\Delta T = 20^{\circ}F$ on; $\Delta T = 3^{\circ}F$ off) implementationg 0 percent or 100 percent flow.

(2) A variable differential control limit ($0^{\circ}F < \Delta T < 12^{\circ}F$) implementing a continuously varying flow 0 percent to 100 percent.

The Rho Sigma study concludes that on days of high insolation collector performance is comparable under either control strategy but that on days of low insolation the proportional control achieves better collector performance.

Hawthorne Industries, Inc., also a manufacturer of proportional differential temperature controllers, has measured performance of systems using both approaches (Reference 6). The results of tests on two days indicate increased energy collection with a proportional controller, particularly on a day of low insolation.

For high temperature collectors, the issue has been addressed analytically at Purdue University using computer modeling and simulation tools. Their conclusions: "... the constant outlet temperature collector is as efficient in collecting solar energy as a constant mass flow rate collector." (Reference 7).

Collector performance simulations done at the Honeywell Energy Resources Center suggest that selection of appropriate fixed ΔT limits for on/off control is the key in comparing performance on both high- and low-level insolation days (Table 1 and Figure 1).

Clearly the issue has not been settled. Research under way at various facilities in the United States is addressing the question of adaptive control for solar heating and cooling systems (References 2, 3 and 8).

Colorado State University is at present conducting experiments with its hydronic Solar House I system to address the issue of proportional versus on/off control in the collector loop (Reference 9).

As part of this control study the issue will also be addressed at the system level, using Honeywell's analytical simulation tools. In approaching the question analytically, an implicit assumption exists that the collector absorber plate efficiency remains unaffected by change in flow. In-house work at Honeywell indicates

Table 1. Simulated Performance of Lennox Two-Cover Flat Plate Collector (Two Meters by One Meter)

INSOLATION LEVEL	TYPE OF CONTROL	ΔT_{ON} ^{o}F	ΔT_{OFF} ^{o}F	ENERGY STORED OVER ONE DAY (W-HRS)	DIFF. %
HIGH - CLEAR DAY MAX. 945 w/m^2 (300 BTU/HR-FT^2)	PROPORTIONAL	3	0	4195	--
	ON-OFF	18	3	4182	-0.3
	ON-OFF	6	2	4253	+1.4
LOW-HAZY DAY MAX. 473 w/m^2 (150 BTU/HR-FT^2)	PROPORTIONAL	3	0	1648	--
	ON-OFF	18	3	1626	-1.3
	ON-OFF	6	2	1726	+4.7

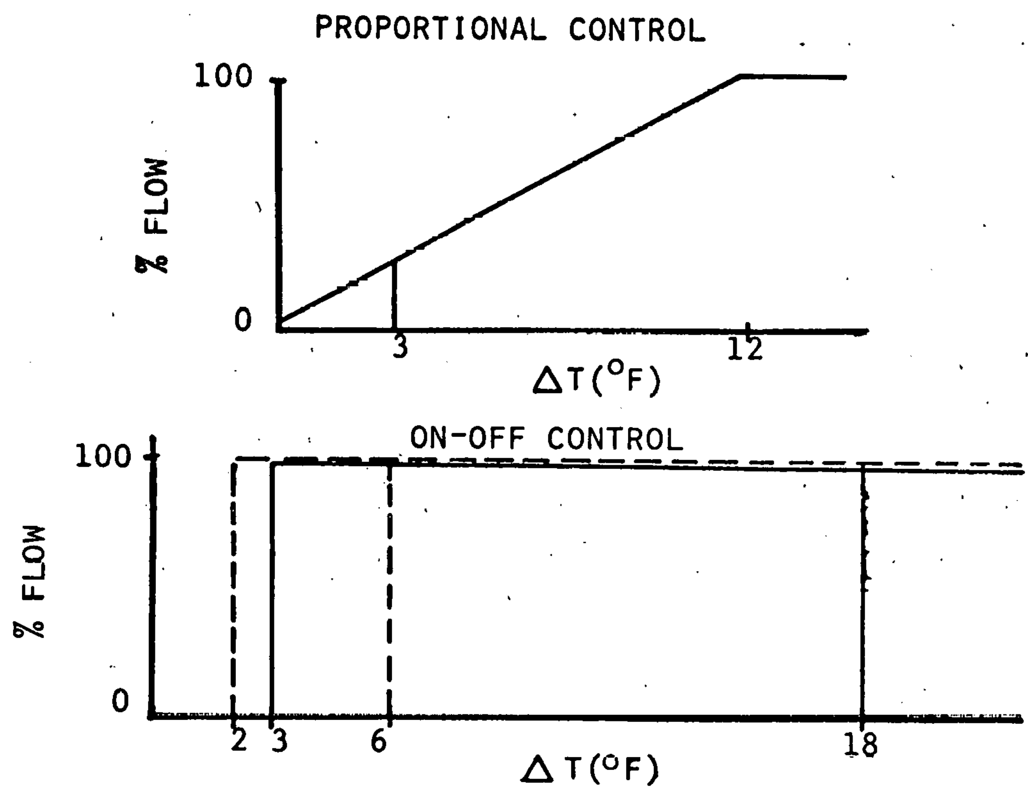

Figure 1. Proportional and On/Off Control Collector Loop Algorithms

that this assumption is approximately correct for flat plate collectors. Controlled experiments were conducted at the Energy Resources Center test facility using a solar simulator. Collector performance was evaluated under varying flow rates for an air collector (Reference 10) and also for a hydronic collector currently on the market. Results reveal that plate efficiency is a weak function of mass flow across the collector (Figures 2 and 3).

Control of ambient air intake, either proportional or on/off, has already proved to be an energy saving technique in both large and small applications. This "economizer" control mode relies on enthalpy and temperature differences between indoor and outdoor air to reduce the cooling load.

2. Economizer Control

In recent years Honeywell has developed an analog computer model in which the thermal characteristics of a house and its HVAC system are represented by electrical resistance circuits. This approach to the simulation of HVAC system performance, with hourly weather data inputs, has been well validated (Reference 11). It has been used extensively to evaluate the energy savings potential of various strategies applicable to single-family residences, including "economizer" control. These simulations indicate that in a single-family residence the reduction in cooling compressor hours due to the cooling effect of ambient air circulation is substantial (Reference 12):

- Fourteen percent in Atlanta,
- Thirty-one percent in Chicago,
- Twenty-eight percent in Minneapolis,

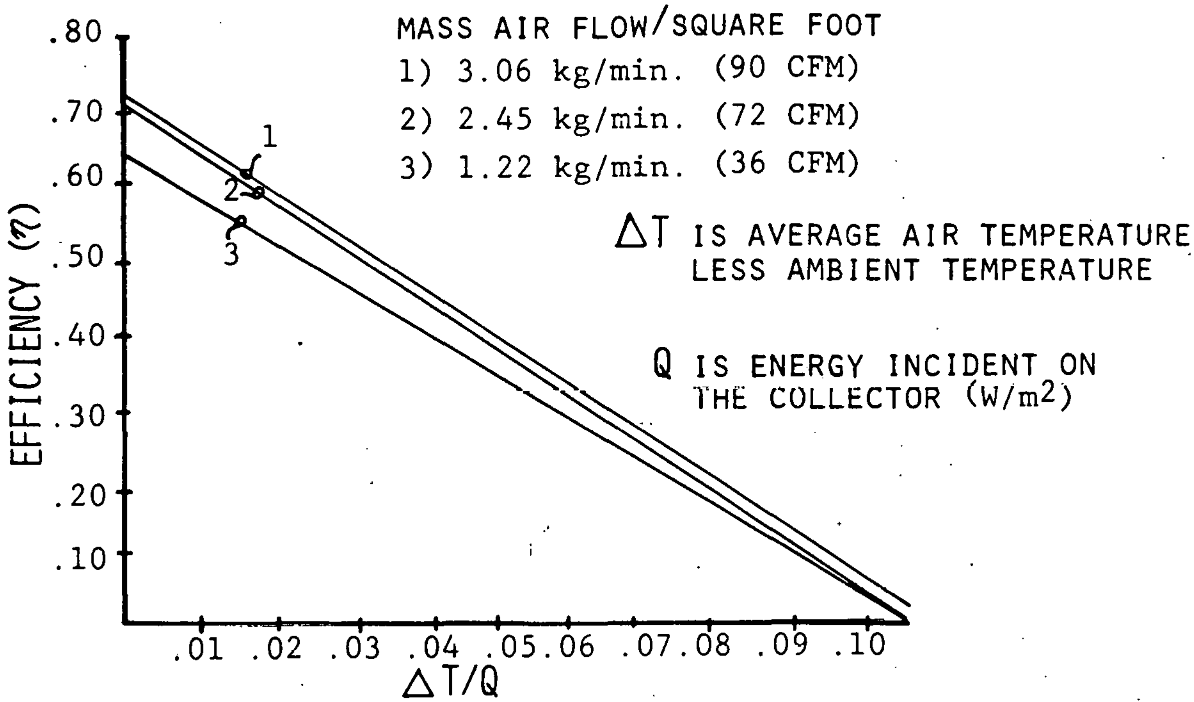

Figure 2. Parallel Plate Air Collector Efficiency

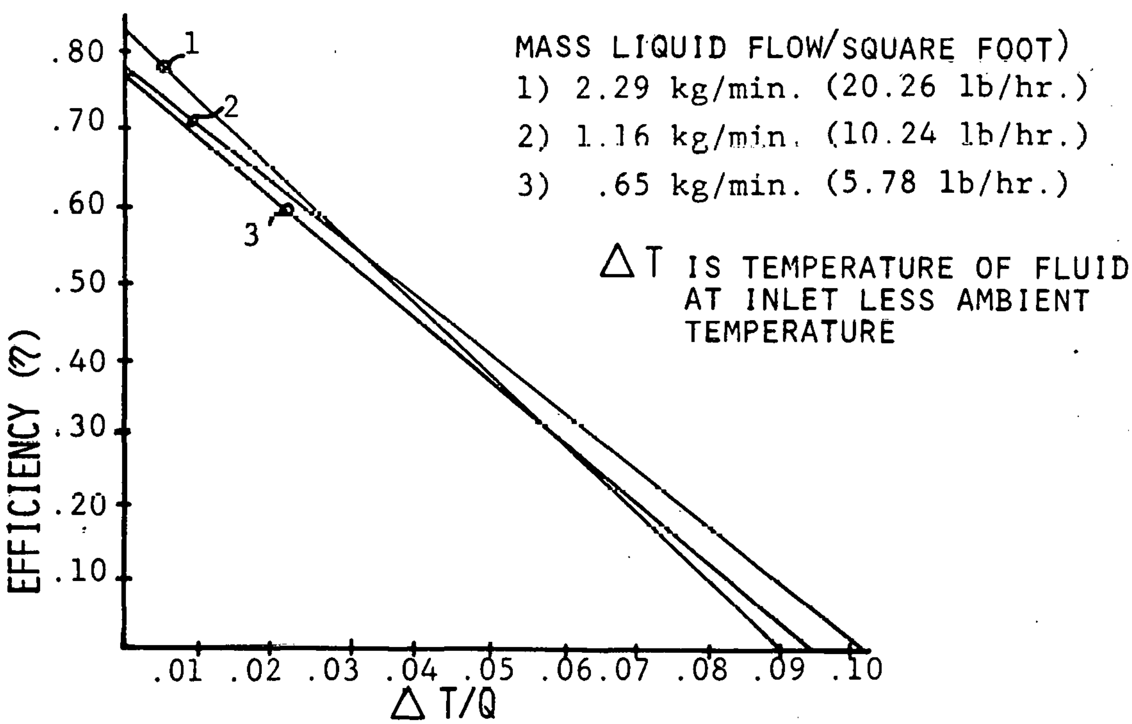

Figure 3. Flat Plate Hydronic Collector Efficiency

- Eight percent in Dallas,
- Thirty-one percent in Pittsburgh,
- Fifteen percent in St. Louis,
- Sixty-one percent in Seattle.

Enthalpy control has already been applied very successfully in two southern California single-family residential solar HVAC systems--Southern California Gas Company Minimum Energy Dwellings--where the days are often warm and dry while the nights are cool and damp.

Due to the higher interior cooling loads generated in most commercial buildings, proportionally greater savings can be achieved with enthalpy (or economizer) control than in single-family residences. Analog computer simulations of a typical commercial building cooling system, with the intake of ambient air controlled, yield the following typical savings in cooling compressor operating hours (Reference 13):

- Eighteen percent in Atlanta,
- Twenty-seven percent in Chicago,
- Seventeen percent in Houston,
- Sixty-eight percent in Los Angeles,
- Twenty-nine percent in Minneapolis,
- Thirty-seven percent in New York,

- Sixty-four percent in Seattle,
- Twenty-four percent in St. Louis.

JRB Associates has developed its own computer simulation, using the ASHRAE "Bin Method," nad has independently generated comparable levels of cooling savings with enthalpy control in commercial buildings (Reference 14):

- Twenty-three percent in Minneapolis,
- Twenty-five percent in Springfield, MA,
- Twenty percent in Washington, D.C.,
- Fifteen percent in Atlanta,
- Thirteen percent in Shreveport, LA.

Even more dramatic results are predicted for one typical building on the University of California, San Diego campus, equipped originally with a fixed ambient intake reheat air-handling system. In this building and climate, control of the enthalpy through variable ambient air intake is expected to save 11 billion heating Btus and 13 billion cooling Btus each year (Reference 15).

It is clear that this type of economizer cooling should be included in all systems and locations covered by this study.

3. Direct Heating Mode

As with proportional control in solar systems, the issue of a direct heating mode in solar system design has also seen the development of two schools of thought. Two engineering camps seem to exist: one that designs systems which include a direct heating mode (through a heat exchanger in the collector loop) and one that designs systems which exclude such a possibility. The tradeoff appears to be between the additional pump and valves required for a direct heating option and the additional thermal losses introduced in storage. Honeywell in-house analysis studies indicate that the additional energy delivered does justify the additional actuators required and the more complex control. As informal survey of the industry indicates that this evaluation is not universally accepted. However to date this survey has failed to uncover published material that speaks to this issue. The question of a direct cooling mode has been partially addressed in experimental work with Colorado State University Solar House I. However results are inconclusive (Reference 16). At the Lawrence Berkeley Laboratories (LBL) a system has been designed and fabricated that does include both direct solar heating and cooling modes (Reference 17). Forthcoming results of tradeoff studies at LBL are expected to include evaluations of these modes.

Because definitive experimental data are lacking at present, analysis and tradeoff evaluations will be included in this study for one of the residential solar systems.

4. Heat Pump Options

The re-emergence of heat pumps as effective devices for space heating and cooling has occurred simultaneously with the arrival

of solar collector climate control systems. These simultaneous developments introduce control options that are both interesting and confounding.

Two types of heat pumps are currently being sold commercially, independent of solar energy considerations:

- An air-to-air heat pump, which extracts heat from outdoor air and transfers it to the indoor heated space. This same heat pump can reverse the heat transfer direction for cooling season operation.

- A liquid-to-air heat pump, which extracts heat from a liquid heat source and transfers it to the indoor heating space. This heat pump can also reverse the heat transfer for the cooling season.

A number of recent studies have evaluated heat pump configurations within a solar augmented heating system. The significance of these studies is found in the lack of consensus as to conclusions.

For instance a detailed computer analysis of air-to-air heat pump systems operation in a northern climate was recently completed at the Engineering Research Institute in Ames, Iowa (Reference 18). Evaluated were three residential configurations using a four-ton heat pump.

- Parallel - Space heating is provided either directly from the solar storage tank or from the air-to-air heat pump.

- Series - Space heating is provided by the air-to-air heat pump. Heat from the storage tank is added to the evaporator inlet of the heat pump to provide a better coefficient of performance (COP) in cold weather.

- Series-Parallel - Either parallel or series operation is provided.

Results of this study indicate that the parallel arrangement results in the highest seasonal performance factor and lowest annual energy consumption. However, the more complex series-parallel arrangement results in a significant reduction of utility peak loads.

On the other hand, a study completed at Brookhaven National Laboratory (Reference 19) concludes that a properly designed series system can achieve lower cost and higher performance than other configurations.

At the University of Wisconsin a third type heat pump has been evaluated. It has two evaporators, one in the storage loop and another outdoors. Results of performance evaluations comparing this dual-source heat pump and an air-to-air heat pump parallel to the solar subsystem favor the parallel arrangement. The additional electricity demands of the dual-source heat pump overcome the auxiliary heat savings generated (Reference 20).

Simulation work at Honeywell using the SUNSIM software indicates that a two-speed dual-source heat pump is prefereable to an air-source heat pump. The results of a year's simulation in Omaha, Nebraska, are presented in Table 2.

All of the studies cited above emphasize the importance of the heat pump COP design profile, the collector-storage sizing and the control algorithms and set points. The thrust of this current contract work will be the design of optimal control algorithms to maximize performance of each system configuration.

Table 2. Single-Family Residence, Omaha, Nebraska

COLLECTOR AREA 504 FT^2 STORAGE TANK 750 GALLONS	SOLAR ASSISTED HEATING SYSTEM AIR-TO-AIR HEAT PUMP	SOLAR ASSISTED HEATING SYSTEM DUAL-SOURCE HEAT PUMP
ANNUAL HEATING LOAD	79.7 BTUS X 10^6	79.7 BTUS X 10^6
SOLAR CONTRIBUTION DIRECT/STORAGE	35.5 BTUS X 10^6	15.8 BTUS X 10^6
HEAT PUMP AIR-TO-AIR	39.5 BTUS X 10^6	25.3 BTUS X 10^6
SOLAR STORAGE	---	35.3 BTUS X 10^6
ELECTRIC HP	20.6 BTUS X 10^6	20.5 BTUS X 10^6
AUXILIARY ELECTRIC HEAT	4.7 BTUS X 10^6	3.3 BTUS X 10^6
TOTAL ELECTRIC INPUT	25.3 BTUS X 10^6	23.8 BTUS X 10^6

SUNSIM SIMULATION

5. Thermostat Setback and Setup Strategies

Single-Family Residences--An interesting question was raised by two research scientists early in 1977 regarding night setback in the heating season. Dr. M.P. Zabinski and Dr. J.Y. Parlanga challenged the general conclusion of a number of papers published since 1973 (Reference 21). The crux of their challenge was that without detailed knowledge concerning the time constants associated with a residential heating system and the heated space, energy savings could not be determined from simulation studies. In fact, they conclude, given a low heating system efficiency and a large time constant associated with the building, nighttime setback might result in negative energy savings. These conclusions appeared to be in stark contrast to earlier work done at Honeywell (Reference 22), at Oak Ridge National Laboratory (Reference 23), and at a number of other research laboratories. In work recently completed at Honeywell

a new look has been taken at the question of nighttime setback in a single family dwelling as it relates to type of system, type of fuel, sizing of furnace, building insulation and location (Reference 24). The tool in this analysis work was Honeywell's well validated and well documented analog-digital hybrid computer simulation (Reference 25). The results are sufficiently important to be stated here in some detail. In gas-fired, forced-air heating systems,

- For effective savings with night setback, the furnace should be sized to achieve temperature pickup in a reasonable amount of time.

- The greater the duration of setback and the greater the amount of setback, the greater the energy savings.

- Savings in gas consumption with night setback are greatest in cold climates (Table 3).

Oversizing the furnace by 20 percent yields a reasonable pickup time from an eight-hour 5° night setback in a cold climate--less than one-half hour when the outdoor temperature is at 38°F and just over one hour when the outdoor temperature is at 0°F. For larger setback differences--10°F, for example--the furnace should be oversized accordingly.

In oil-fired warm-air heating systems,

- Setback savings should be comparable to gas-fired systems.

In central electric warm-air furnace systems,

- Setback savings should be equal to or slightly higher than those savings shown for gas.

Table 3. Energy Savings with Night Setback for a Single-Family Residence, Gas-Fired Warm Air System

LOCATION	CONSTRUCTION	NIGHT SETBACK FROM 70°F	ANNUAL GAS INPUT BTUS X 10^6	ANNUAL GAS SAVED BTUS X 10^6	ANNUAL BLOWER HOURS	ANNUAL HOURS SAVED
MINNEAPOLIS 8270 DEGREE DAYS	WALL INS. 3-5/8 INCHES CEILING INS. 6 INCHES	NONE	61.27	--	1765	--
		5°F	58.12	3.15	1580	10.5
		10°F	56.04	5.23	1450	17.8
CHICAGO 6067 DEGREE DAYS	WALL INS. 3-5/8 INCHES CEILING INS. 6 INCHES	NONE	38.18	--	1244	--
		5°F	35.41	2.77	1070	14.0
		10°F	34.19	3.99	988	20.6
ST. LOUIS 4837 DEGREE DAYS	WALL INS. 3-5/8 INCHES CEILING INS. 6 INCHES	NONE	27.89	--	937	--
		5°F	25.53	2.36	785	16.2
		10°F	24.82	3.07	727	22.4
ATLANTA 2924 DEGREE DAYS	WALL INS. NONE CEILING INS. 2 INCHES	NONE	26.35	--	859	--
		5°F	23.79	2.56	700	18.5
		10°F	22.36	3.99	623	27.5

ALL FURNACES OVERSIZED BY 60 PERCENT

In hydronic heating systems,

- Setback savings would be comparable to those shown for gas warm-air systems if the boiler and radiators are sized in proportion to the design heat loss.

- If the boiler capacity exceeds radiator output, setback savings would be less than for gas warm-air systems.

- With boiler control that sets the boiler temperature as a linear function of outdoor temperature, setback savings would be comparable to those shown for gas warm-air systems.

In an air-to-air heat pump system,

- Setback savings will be less than the savings shown for gas warm-air systems.

Dual setback strategies, applicable when all family members are away for long daytime periods, have also been evaluated. As expected, additional energy savings are generated with dual setback over sufficiently long periods.

With the same hybrid computer model and simulation program cited previously, two energy saving <u>cooling</u> strategies applicable to single-family residences have also been evaluated at Honeywell (Reference 26):

- Daytime thermostat setup, and

- Nighttime thermostat setup.

Table 4 presents the results of simulations for three cities representing three different cooling climates in the United States. These results indicate that setup strategies do indeed save some energy in the cooling season. However, the applicability of such strategies is dependent upon occupancy and also upon the size of the air conditioning unit. Where family members are away from home during the day, cooling season setup strategies are of value. The rate of recovery from setup is a function of air conditioner size vis-a-vis the building response. For the single-family residential control systems to be designed and evaluated in this study, only setback heating strategies will be included.

Table 4. Cooling Requirements for a Single-Family Residence with Central Air Conditioning

LOCATION	COMPRESSOR DESIGN SIZE	SETPOINT 78°F	SETPOINT 82°F (8AM-4PM) 78°F (4PM-8AM)	COMPRESSOR HOURS SAVED
MINNEAPOLIS 1956	20,535 BTU/H	434 HOURS	411 HOURS	23 HOURS
ST. LOUIS 1957	22,247 BTU/H	928 HOURS	869 HOURS	59 HOURS
DALLAS 1952	27,709 BTU/H	1187 HOURS	1153 HOURS	34 HOURS

Commercial Buildings--In view of the results of studies relating to single-family residences (Tables 3 and 4), it is apparent that night setback in the heating season and night setup in the cooling season are energy saving strategies. In the case of a small commercial building system, such as the one to be evaluated in this study, building dynamics and occupancy patterns make both of these strategies good candidates within the design of cost-effective control systems.

For the large commercial building system included in the study, the hybrid simulation results are really not applicable. However, beginning in 1973, a set of strategies has been applied to Honeywell's existing large Corporate office building that have had dramatic results in energy savings since that time:

- Comfort range temperatures of 68°F for heating and 78°F for cooling.

- Operating hours reduced from 20.5 to 11.5 hours, including janitorial services.

- The heating/cooling system and all lights turned off at night 12 months of the year.

These strategies, plus some system and building improvements, have resulted in a 50 percent reduction in energy requirements since 1973. Therefore they will be included as strategies in the large commercial building system to be evaluated in this study.

6. Temperature Reset Strategies

The term "reset" is used to group a number of system strategies that provide options other than "absolute" temperature control. For example, a temperature control setpoint might be a function of time, or a function of some other temperature setpoint, or a function of a temperature sensor reading.

In collector loop operation, the "reset" term is applied to the control when the temperature setpoint is a function of the storage tank temperature. In most state-of-the-art solar hydronic systems this is indeed the case--the collector loop operating setpoint might be anywhere between 3°F and 20°F above the storage tank temperature, depending on the differential temperature controller that is used. However, in a direct heating or cooling mode, the collector loop is usually under "absolute" temperature control--the operating temperature setpoint in the loop is fixed. Inasmuch as thermostat setback is rapidly becoming commonplace, it would seem that differential temperature control of the collector loop, using the sensor in the room thermostat as the comparator, should provide additional energy for direct heating. Differential temperature control in solar air heater systems is already standard.

In storage tank operation there are two reset options that should be evaluated at the system level. These options determine the operating temperature setpoint for heating from the storage tank.

- Differential temperature control of the storage tank using the room temperature as the comparator.

- Storage tank temperature control as a function of outdoor temperature.

The second option depends on an outdoor sensor and a functional algorithm to determine the temperature setpoint. A likely candidate for the Option 2 algorithm prescribes the storage tank operating temperature setpoint (T_{min}) as follows:

$$T_{min} = 70^{o}F + (70^{o} - T_a)\,\frac{2}{3}$$

where T_a represents the temperature in the outdoor sensor. Such an algorithm should be implemented with constant furnace fan operation in a forced-air system to maximize comfort.

A tradeoff evaluation between absolute temperature control and any of the reset options must include the constant blower fan power required and the cost of the control units.

In a solar air conditioning system, storage loop reset algorithms become particularly important. For instance, in an absorption cooling system, the inlet temperature to the generator determines the COP of the system as well as its operating capacity. If the absorption unit is in series with the solar collectors, an effective algorithm should minimize the temperature at the outlet of the generator to maximize collector efficiency, yet at the same

time ensure a temperature at the inlet of the generator sufficiently high to provide adequate cooling capacity. This means that operating strategies should encourage lower COP's when the system is operating at part load (Reference 27).

In the absorption cooling system tradeoff studies, system performance with an ARKLA unit will be evaluated. Fixed inlet temperature setpoints will be measured against a temperature reset algorithm:

$$T_{min} = 175^{o} + 3/2\ (T_{ctw} - 80^{o}F)$$

where T_{min} represents a minimum temperature at the inlet of the absorption unit generator and T_{ctw} the water temperature in the cooling tower, or an approximation by ambient temperature. Once this minimum temperature has been determined, a boiler or heating element in the storage loop can be activated if required.

7. Off Peak Strategies

Two recent developments in energy conservation, independent but apparently synergistic, are creating a climate for new off peak strategies.

Historically the power industry has attempted to build power capabilities to satisfy the peak demands of its customers, leaving excess energy available during the longer off peak periods. Recently, however, awareness of energy limitations has led to efforts directed at equalizing demand periods.

The development of solar energy as a practicable energy source has resulted in climate control systems that include large storage

masses, such as liquid filled tanks and pebble filled bins, to store excess solar heat for later use.

These twin developments provide the setting for imaginative strategies that might minimize user peaks for the hours and days ahead. Some strategies are geared to load management, others to storage management. Some are heating strategies, others are for cooling. But all have in common some element of weather anticipation. Within the framework of this survey, it seems appropriate to address the questions of weather anticipation and storage management as they relate to both heating and cooling systems.

Weather Anticipation--From a literature search it appears that there is a wide variety of sophistication among the algorithms that have been proposed for weather anticipation.

An example of a very simple, yet effective, approach is found in the strategies defined to control the heating and cooling system for the 36,000 square foot conventional all-electric Rauenhorst office building in Minnesota (Reference 28). Two 40,000-gallon underground insulated tanks provide the storage capacity for either heating or cooling during periods of peak power demand. The simple off peak strategies are manually activated:

- Summer (4 months), two cooling tanks;
- Fall (2 months), one cooling tank, one heating tank;
- Winter (4 months), two heating tanks;
- Spring (2 months), one cooling tank, one heating tank.

Somewhat more refined is the weather model used at the University of Wisconsin to evaluate design tradeoffs for a residential solar air heater system (Reference 29). The system analyses were based on computer simulations using hourly meteorological data for a typical year. The off peak storage management strategies, however, follow a load model based on two monthly parameters that are available for most metropolitan areas from the National Climatic Center:

- The normal daily minimum temperature for the month, and
- The record lowest temperature for the month.

It is expected that this load model will be incorporated into the control system when it is eventually implemented on an experimental solar home constructed by the University Solar Energy Laboratory (Reference 30).

Somewhat more sophisticated is a weather model developed for cooling season load forecasting by the Virginia Electric and Power Company (Reference 31). The Vepco model was developed by correlating U.S. Weather Bureau data over six years. The model predicts the summer afternoon peak weather load as a function of the actual high and low temperatures for the three preceeding days and the forecasted afternoon high temperature, dew point, cloud cover and wind speed. A comparison of the weather model's predicted peak and the actual peak for five continuous weeks in July and August at Richmond, Virginia, yields close correlation: an average error of 1.9 percent and a standard error of 2.47 percent.

Power Technologies, Inc., of Schenectedy, New York, has developed a weather model that forecasts a 24-hour load curve up to seven days ahead (Reference 32). The capabilities of this computer pro-

gram are probably well beyond the requirements of peak load forecasting as it applies to off peak strategies for a solar heating or cooling system.

A limited amount of work appears to have been done in the prediction of possible sunshine. At least in some locations the short term prediction of possible sunshine during the heating season may have some merit. For instance, in Minneapolis there is a reasonable probability (68 percent) that the fraction of possible sunshine for two consecutive winter days will not differ by more than 40 percent (Reference 32). Because a substantial amount of statistical work must be done to develop any kind of sunshine prediction pattern, the control strategies to be included in this study will include temperature anticipation only.

<u>Storage Management</u>--It is in this area that the development of appropriate strategies may yield substantial reductions in peak load demand and savings in user dollars. In a residential heating system, storage management may mean simply charging the storage tank during the off peak period to a predetermined minimum temperature prior to the peak avoidance period. In the Rauenhorst conventional system cited previously, the temperature selected for the entire heating season is 200°F (Reference 28). The algorithm used in the University of Wisconsin solar evaluation is also fairly simple and conservative. During each off peak period the storage tank is charged to some minimum temperature (T_{min}) that will assure a minimum energy availability (Q_{min}) for the peak avoidance period ahead (Reference 29).

$$Q_{min} = UA\,(T_{set} - T_{LD})\ (24 - \text{number of off peak hours})$$

$$T_{min} = Q_{min}\,/\overline{m}c + T_{mu}; \text{ and}$$

$T_{mu} = UA\ (T_{set} - T_{LH})/\dot{m}c + T_{set}$;

where UA is the building design load,

$\overline{mc}$ is the thermal capacitance of the storage mass,

T_{set} is the room thermostat set temperature,

T_{LD} is the normal daily minimum temperature by month,

T_{mu} is the minimum useful temperature, and

T_{LH} is the record lowest temperature for the month.

More complex and perhaps more interesting is the management of dual storage within a solar heating or cooling system to minimize electrical usage at high daytime rates and to achieve temperatures that are appropriate for efficient system operation.

To date there seems to be little motivation for peak avoidance strategies in solar systems where the auxiliary energy source is natural gas. However it is a fact that during peak demand periods, utilities depending upon a fixed rate of natural gas distribution find it necessary to add propane gas and air into the local consumer gas. This mixing, at extra cost to the utility company, may portend a change in rate schedules at some future time.

8. Peak Load Strategies

In any system, solar or conventional, there are groups of strategies to avoid or reduce energy usage during designated peak periods. These strategies are most critical and therefore most effective in

large plants, but they also have meaning in small applications. Load shedding in a solar heating system may well translate to a heat from storage mode during certain hours with a "timer" built into the control system. In a large commercial system designed for heating and cooling, load shedding may call for chilling from the cold storage tank as well as the usual reduction in lighting and other heat-producing appliances. Duty cycling of the furnace or chiller on a time schedule or as a function of extreme outdoor temperatures has application in solar as well as conventional heating and cooling systems. In heat pump systems these two groups of strategies combine in ways that can be very interesting--for instance, the sequenced duty cycling of several heat pumps in a large zoned building.

B. CONTROL IMPLEMENTATION

Judging from the current market literature "advanced solid state electronics" are state-of-the-art in the control of solar heating and cooling systems.

Closer examination, however, reveals that controllers being marketed present a wide spread in the mulitiplicity and flexibility of their functions.

For the purpose of this survey, advanced means electronic relative to sensors, while advanced controllers include only those that are microprocessor based. The discussion is delineated by four groupings:

- Sensors,
- State-of-the-art controllers,

- State-of-the-art multiple control modules,
- Advanced microprocessor-based control systems.

1. Sensors

Rapid changes in the field of control implementation are reflected in developments related to the types of sensors now appearing on the market.

Temperature sensors find multiple applications in most solar climate control systems. It is true that some collector types are more amenable to flux measurements than to temperature measurements. However since the main thrust here is control of flat plate systems, emphasis will be on developments relating to temperature sensors. Commercially mature sensors now on the market generally are of the following three types.

Electromechanical Sensors--Expansion of heat-sensitive fluid in the sensing bulb causes a switch to close, activating or deactivating the appropriate actuators. These are often used in locations where the sensor is remote from the control unit. Note that electromechanical humidity sensors achieve the same switching result using a moisture sensitive element such as a nylon ribbon.

Bimetal sensors are most commonly used in room thermostats. Two dissimilar metals are laminated together to produce a bimetal coil. On a rise in air temperature the coil unwinds sufficiently to close a mercury switch, activating or deactivating the appropriate actuators.

Electronic Sensors--There are two types of electronic sensors on the market that relate to the HVAC applications in this study. The more familiar is the thermocouple, which consists of two dissimilar metals coupled together at two junctions. A current will flow in the resulting circuit proportional to the difference in temperature between the two junctions.

A thermistor, on the other hand, is essentially a semiconductor composed of a mixture of metallic oxides and silicon. The thermistor behaves like a thermal resistor with a high negative temperature coefficient of resistance. The fluctuating output voltage resulting from either a thermocouple or a thermistor response can be an analog signal proportional to the temperature change or a high/low digital signal, depending on the control design requirements. In high temperature applications the electronic thermocouple sensors (chrome/aluminum or iron/copper) and the very accurate platinum resistance temperature devices (RTD) are frequently found. However in applications where electromechanical sensors are now at home, thermistors--silicon based semiconductors--are increasingly specified.

Pneumatic Sensors--A bimetal or rod and tube sensing element responds to a change in temperature by modifying the pressure output of the sensor. This pressure difference is amplified by the local controller to effect a pressure change in the branch line to the actuators. Not suprisingly pneumatic sensors are found in the larger pneumatic systems.

This survey reveals that electromechanical sensors are still widely used in the state-of-the-art control systems defined previously. To the extent that hysteresis qualities are important--such as in differential temperature control where actuator cycling is to be

avoided--electromechanical sensors are preferred. However in monitoring devices and complex systems requiring precise temperature control, hysteresis qualities can introduce problems.

Electronic thermistors already display qualities that find application in state-of-the-art control systems as well as in advanced microprocessor based systems. They are smaller and therefore more amenable to tight installation requirements than are their electromechanical counterparts. Lacking in hysteresis qualities, they are nevertheless adaptable to hardware and software modifications that can supply these missing qualities. Given the superior durability and the modest materials content of electronic sensors, they will no doubt become more commonplace as costs continue to decrease.

In control of the collector loop, the placement of sensors has some significance. Ideally, the sensor should measure the temperature of the heat transfer medium at the outlet of the collector. However there are practical problems inherent in this--maintenance, tapping the line and sealing. Therefore an alternate approach is often followed. The sensor is mounted on the collector plate at a position that will accurately predict the temperature of the fluid at the outlet of the collector. At Honeywell's solar laboratory, tests of one collector type have shown that the plate temperature approximates the fluid outlet temperature at normal operative temperatures when the sensor is pressed against the absorber tube about one-third of the distance from the inlet. Of course, the optimum location will vary with the type and make of the collector.

2. State-of-the-Art Controllers

Differential Temperature Controllers--There are available on the market a variety of differential temperature controllers, differential temperature control being basic to the operation of most solar climate control systems. At the minimum these controllers activate a single pump or blower of fixed or variable speed. A few of the controllers have two or three poles to control one or two additional functions.

Most units have the additional features, often optional, of high and low collector temperature protection. Typically the high temperature feature is a line voltage output to control the pump or blower. The low temperature feature, commonly called "frost cycle," can be line or low voltage, controlling a pump, valves, or auxiliary heating element.

These controllers generally are sold with factory-set differential temperature limits--a higher "delta T" to switch on and a lower "delta T" to switch off.

Below is a partial list of manufacturers who market these controls.

- British American & Eastern Company, Inc.
- Del Sol Company
- Energy Converters, Inc.
- Hawthorne Industries, Inc.
- Heliotrope General
- Honeywell Inc.
- Independent Energy, Inc.
- Johnson/Penn Controls
- Natural Power, Inc.
- Pak-Tronics
- Rho Sigma, Inc.
- Robertshaw
- Solar Control Corporation
- Solaray, Inc.
- Sol Stat Controls
- Sunwater
- Troger Enterprises (Voltaic Sensor Control)

Inasmuch as various manufacturers' prices represent different methods of marketing, it is difficult to make direct price comparisons.

Nevertheless it may be helpful to note that the list prices vary from a minimum introductory offer of $29 to $148 for a unit that contains most of the functions described above.

Since "proportional" control is an issue in this survey, it is important to address this concept as it relates to conventional differential temperature control units. Rho Sigma markets models that are representative of two approaches: Model RS142 provides two-stage control of a two-speed pump or valve; Model RS500 provides continuous variable control of a single-speed pump by pulsing the voltage at a frequency that produces a flow proportional to the temperature differential.

Hawthorne Industries' proportional controller also controls a single-speed pump by suppressing a portion of the phase in each wave of electric current. The result is a smooth variable transition in speed control.

At wholesale the cost differentials for proportional control vary from $5.00 (Hawthorne Industries, Inc.) to $7.00 (Solar Energy Products, Inc.) and $7.75 (Rho Sigma, Inc.).

For even the simplest residential or small commercial solar augmented heating system, the control units under discussion must be augmented by relays, manual switches, lights, transformer and a suitable panel on which to mount and interconnect the individual components. The costs for such a panel with enclosure are not insignificant. For instance, Honeywell has been involved in several demonstration projects supported by HUD. Cost for the

control panel and box, including arrangement and assembly of the control components, has varied from \$527 to \$1000 for these single-family heating system applications. Of course, to this cost must be added that of all the units mounted on the panel.

In a large commercial solar heating and cooling installation, the number of control units, relays and switches expands accordingly.

<u>Thermostats</u>--Thermostat control of space heating systems, using a temperature sensing element, dates back to the early years of this century: a change in temperature triggers a relay that either activates or deactivates some actuating hardware component. Over the years anticipators--usually tiny heating elements--have been added to minimize fluctuations in the space temperature. As early as the 1930's thermostats with timers were on the market to achieve automatic thermostat setback for some period within the 24-hour day. By the 1950's, however, the seemingly unlimited supply of cheap fuel rendered these advanced thermostats cost-ineffective for most applications.

The awareness of an energy crisis in the early 1970's produced several distinct HVAC developments that interact with each other as well as with the thermostat controls:

- Energy conservation setback and setup strategies,
- Economizer cooling,
- Solar collector systems,
- Heat pump systems.

In the large commercial energy management systems on the market today, thermostat functions involving any or all of these can usually be either hardwired or programmed into the control systems.

However it is not yet clear how manufacturers of residential HVAC controls will respond to this multiplicity of functions. Indeed there are at present controls on the market responding adequately to each individual development.

A recent survey lists seven major manufacturers marketing energy saving reset thermostats (Reference 33). Humidistat-controlled economizers that provide fresh air for cooling purposes are now state-of-the-art in the design of virtually all air conditioning systems.

Two-stage thermostats, sequencing alternate sources of energy, are useful in any of several applications: in solar heating systems with conventional fuel as backup; in economizer cooling systems with any of several air conditioning systems as backup; and in heat pump systems with electric resistance backup heating.

It remains to be seen whether improvements in conventional thermostats can indeed handle the multiple functions demanded by solar heat pump systems that also incorporate economizer cooling and automatic thermostat reset functions. It appears that at some level of complexity an integrated circuit will be preferred.

Electronic thermostats may be just two or three years away from the marketplace. Predictions vary, but it seems likely that a user programmable thermostat including various timed functions will soon be available ($100 in today's dollars).

Reset Controllers--Whenever a large storage mass is incorporated into an HVAC system, a "useful" temperature, or setpoint, of the mass must be defined. In simple control this setpoint may be absolute, as it is in several of the baseline HVAC systems defined for this study.

Reset controls are now being marketed at relatively low cost to vary the storage setpoint according to the demands that are likely to be placed upon it. Since the demands, heating or cooling, are a strong function of outdoor temperature, the remote sensor associated with the reset controller is mounted to sense this temperature. Honeywell's reset controller has four options in selection of the ratio of reset delta to sensed temperature. Natural Power, Inc. of New Hampshire is marketing a reset controller in which this ratio is user adjustable. Both of these reset controllers list at under $150.

3. State-of-the-Art Multiple Control Modules

In view of the multiplicity of functions involved in even the simplest solar climate control systems, there is a current trend toward packaging solar control systems into modules.

Honeywell has recently added to its residential controls line a Solar Control Panel, Model W968A. This panel contains a differential temperature controller, transformer and switching relays that can control either a three-pump (single-speed)/two-valve or a two-pump/four-valve solar hydronic heating system. An optional feature is a set of lights on the cover to indicate mode of operation. The minimum collector operating setpoint is controlled by a potentiometer on the panel. The factory-set differential control

limits may be adjusted by changing the plug-in resistors and sensor connections. The entire control panel is 13-9/16 in. x 10-5/8 in. x 6 in. (344.5 mm x 269.9 mm x 152.4 mm).

The Honeywell Solar Control Panel W968A with mode indicator lights has a tentative base price of less than $500.

A modified version of this module to control a parallel solar and air-to-air heat pump system is expected to be announced shortly. This control panel will operate three pumps, two valves, heat pump electric power and auxiliary electric. Current development work at Honeywell Residential Division is expected to result in modules designed for single-family air collector heating systems, with gas, oil, or electric auxiliary backup; for air collectors in parallel with air-to-air heat pump systems; and for hydronic collector systems in parallel with water-to-air heat pump systems.

Costs of these developing systems are expected to be lower, reflecting the benefits of previous work and increased production.

The advantage of a preassembled package for specific applications is substantial. Even lacking a current demand to justify mass production pricing, pilot-production savings are indeed passed on to the consumer. The limitations of this packaged module approach are in the lack of flexibility. The packages are designed for the application specified, and modifications are not practicable.

The solar control manufactured by Natural Power, Inc. also carries a $500 base price, with additional costs for extra sensors and outputs. The controller is designed to control either hydronic or air collector space heating systems with auxiliary furnace backup. The solar control brochure describes the controller as

modular in nature for expanded or contracted functions. The basic unit reads temperatures from six sensors and provides differential and absolute control outputs to a variety of actuators, with setpoints user adjustable.

Solar Control Corporation's Homemaster Solar System Controller is factory programmed to control five different modes of solar and HVAC operation for either a hydronic or an air collector system. The input power channel includes signals from the various sensor probes, from the thermostat and from the temperature setpoints, which are user programmable. The output power channel controls an "unlimited" number of pumps, blowers, valves and dampers involving the collectors, domestic hot water, heating zones and the furnace.

From the literature, it appears that there is system flexibility through easy addition of customized mode hardware. Cost of a unit with standard control logic is $249.

4. Advanced Microprocessor-Based Control Systems

There are a number of programmable microprocessor-based control systems designed and built for specific solar heating and cooling system applications. As an example, the Honeywell Energy Resources Center has designed such a system for the Iowa State University Energy Resources Research House (Reference 34). This control system utilizes a 6800 8-bit microprocessor to control a solar augmented heat pump system for heating and cooling.

Additionally under DOE Contract EG77-C-03-1598, Honeywell is developing a flexible and general purpose controller concept--the Solar Energy Management Controller (SEMS). It can easily be

field-tailored to fit a wide variety of residential solar systems by means of the system initialization procedure, namely:

- Select the system's operational modes;
- Select the desired initiation/termination criteria for each mode;
- Select special situations in which the assigned mode definitions and priorities can be overridden;
- Define relative mode priorities;
- Implicitly define which modes can occur simultaneously and which modes are mutually exclusive;
- Define the required state (energized or de-energized) of solid state relays for each operational mode;
- Enter values to be used for all temperature setpoints, differentials, etc.

At Lawrence Berkeley Laboratories a modest-cost electronic controller has been designed and fabricated. This preprogrammed controller is now being tested and the algorithms evaluated with the LBL experimental heating and cooling system (Reference 35).

NASA-Langley has on its grounds a 1500-square-foot house with solar collectors and a water-to-air heat pump system. This system is presently being controlled by a Tektronix 4051 minicomputer. Control includes temperature schedules for zoned heating and night setback and for cooling (Reference 36).

At least four commercial firms have produced marketable systems for single-family, multifamily and commercial building applications of solar energy.

The Micro-Thermonic Remote Control Unit manufactured by Solar World contains a factory programmed microprocessor. The unit, which lists at just under $1000, handles inputs from up to nine thermistor probes, six temperature reference setpoints and four differential temperature references. The output drivers are electronic switches capable of driving all conventional ac relays. Motorized valves may be driven directly via the electronic switches up to currents of one ampere. The microprocessor design outputs control 11 actuating devices, including collector trackers, various pumps and blowers and auxiliary heat modes.

Rho Sigma, Inc. has on the market the RS600 Programmable Control System. This system utilizes an F8 8-bit microprocessor and includes 16 analog inputs, 16 digital inputs, 16 output channels and a five-digit display. The 16-key keyboard can be used to modify 51 constants in the 1K RAM memory. The main program is stored in the 4K PROM nonvolatile memory. It appears from the literature available that the factory programming of the microprocessor is design specific. The output relays are design selected: 10-ampere solid state switches, 10-ampere contact relays, or 24-volt dc relay drivers. The cost of the RS600, substantially more than $1000 per unit, reflects the individualized specifications of each control unit.

General Electric has available a Solar Energy Heating and Cooling Control System with solid state memory. Programming is flexible according to user requirements so that the unit may be used with any solar heating and cooling system. Memory inputs include four

analog and 16 digital, with 12 output channels, all at an estimated price of $6000.

The Sunkeeper Controller manufactured by the Andover Corporation incorporates solar collector and storage control functions into a large automated building management system currently marketed at $10,000. The controller contains a minicomputer with 32 analog inputs, 32 digital inputs, 32 digital output channels and interface logic for printer or cathode ray tube communication.

It is clear that the cost quotes on these commercialized advanced solar controllers are of fleeting value. Undoubtedly the prices of these units will be affected by design standardization and mass production as dramatically as were the prices of pocket calculators in recent years.

Similarly affected will be the cost of the solid state relays that provide the interface between the microprocessor outputs and the more conventional actuating components in the system. Presently solid state relays cost about twice as much as corresponding electromechanical relays and are many times more reliable provided they are properly installed.

The Singer Company, manufacturer of heat pumps, is currently marketing an electronic control and alarm panel for a four-stage heat pump system. The dimensions of the panel are 24 in. x 18 in. x 6 in. (609.6 mm x 457.2 mm x 152.4 mm). It includes a number of functions worth noting here: four stages of heat rejection, eight stages of supplementary heat control, automatic outdoor reset, adjustable heat setpoint, indicators of water loop and outside air temperatures and high/low temperature alarms. It is assumed here that the Singer Climate Control Division is developing heat pump controllers that also include solar functions, though information on this is not available for this survey.

Honeywell Residential Division expects to have a cost-effective microprocessor-based integrated heat pump system controller on the market for the 1980 heating season. The inputs include analog temperature sensor inputs and a four-stage thermostat generating four analog outputs. One of the special customer-oriented features will be the diagnostic outputs in case of system malfunction. The inclusion of solar, economizer and peak load avoidance functions is not presently considered cost effective. However, if current prognoses are altered due to unanticipated changes in electrical rate schedules or federal legislation, some or all of these functions will be considered.

A number of companies are already well into microprocessor or minicomputer centeral control of larger building energy management systems. Allen-Bradley, Barber Colman, Energy Management Systems of North Carolina, Esterline Electronics, Honeywell, IBM Corporation, Johnson Controls, Powers Regulator Company and Robertshaw have come to the attention of this survey. All of the models documented provide power control through various devices and strategies, and most have either intrinsic or optional HVAC functions such as enthalpy control and chiller and boiler optimization.

Typical power control and HVAC functions found in models of the manufacturers listed include:

- System performance optimization by delaying or accelerating the start of equipment depending on continuous weather data inputs.
- Duty cycling to conserve energy when space conditions are steady state.
- Shedding of low priority electrical loads when electric usage rate inputs indicate peaking.

- Lighting conservation through programmed occupancy times, janitorial schedules and weekend/holiday schedules.

- Use of preferred air source in terms of heat content (economizer cycle) through control of dampers.

- Air conditioning chiller discharge air reset capabilities depending on indoor and outdoor climatic data inputs.

- Optimization of boiler performance through temperature reset based on climatic data inputs.

List prices for the entire central control systems, with the HVAC functions, appear to start at $60,000.

As to how such systems might interface with solar heating and cooling functions, there are two schools of thought:

- Given the modularity of large microprocessor- or minicomputer-based central control systems, it is relatively inexpensive and easy to incorporate the additional solar control functions--inputs, logic and outputs--into the central control system.

- Given the additional memory and input/output channels required for solar system operation, it is better to provide a separate microprocessor control unit for the solar system plus the appropriate interface with the central control system.

Development work under way at the Honeywell Energy Resources Center reflects the second opinion--using local microprocessor control of the collector field that interfaces with the central building control system.

The most recent developments in miniaturization make all of these seemingly advanced microprocessor-based systems already potentially obsolete.

A year and a half ago, a new generation of microprocessors began appearing on the market--the Single Chip Microcomputer. To date several manufacturers are marketing these little items that typically include:

Central Processing Unit (CPU)
1K - 2K Bytes of ROM
64K - 128K Bytes of RAM
24 - 32 I/O Lines
8 - 16 Bit Timer.

Single Chip Microcomputers now on the market include the Mostek MK 3870, Intel 8048, Motorola MC 6801, and Synertek SY 6500/1. Prices are $5.00 to $12.00 per unit in quantities of 5000 or more. It is expected that $5.00 per unit in quantities of 1000 will be standard by the second quarter of 1979.

Clearly these developments are significant in terms of recommendations being made under Part II of this report.

C. CONTROL MECHANIZATION

Within the survey of strategies relating to solar heating and cooling systems, an important issue has been defined that impinges directly upon the selection of actuators, namely: the issue of proportional versus on/off control. A second, more subtle issue has surfaced within the survey of control implementation. It is the issue of central versus local control implementation, particu-

larly in relation to the use of microprocessor-based control systems. Therefore this survey of actuators will concentrate on their suitability for proportional control, their amenability to central control implementation and their applicability to various systems. The discussion will be application delineated:

- Residential and small commercial systems,

- Large commercial systems.

The actuators in a solar heating or cooling system consist of sets of components. In a hydronic system, these components are pumps and valve assemblages; in an air system they become blowers and damper assemblages. However, even the simplest heating or central air conditioning system may be of a hybrid nature. As will be seen from the discussion below, the nature of the heat transfer medium (fluid or air) has little effect on the issues defined above.

1. Residential and Small Commercial Building Systems

Inasmuch as the question of proportional control in the solar loop has already been raised, the question of cost must follow. What cost differential does proportional control introduce in the selection of the loop actuator components? Obviously proportional control can be actuated in two ways. On one side, pumps or blowers with multispeed control or continuously varying control can be selected, while the operators within the valve or damper assemblies remain open/closed or A/B. On the other side, modulating valve or damper operators can be selected, while the pumps or blowers remain constant speed.

The two major manufacturers of proportional differential temperature controllers--Rho Sigma and Hawthorne--rely on variably controlled single-speed pumps as the actuating components. Because pumps generally are tested and rated at their design speed, the question of performance under variable speed control has been raised. Motor failure may become a real problem since motors tend to run hot at very low speeds. Also there may not be sufficient lubrication of self-lubricating bearings. At least one firm, again Rho Sigma, manufactures a differential temperature controller that taps a two-speed pump to vary flow in the solar collector loop.

There are a number of manufacturers offering single-speed and multispeed pumps for solar heating and cooling systems. Typically the taps in multispeed pumps are grouped at the high-speed end of operation so that very low-speed operation is avoided.

Recently Honeywell has done some experimental work regarding the response of a single-speed pump to variable speed control. Results of some of these tests indicate a generally linear relationship between the flowthrough and the power draw of a Grundfos 1/20 hp pump controlled by a commercialized proportional controller. At present there does not seem to be available on the market a pump of sufficient horsepower for a residential space heating system (1/4 HP) that is amenable to this kind of proportional control in the collector loop.

The solar collector loop is generally "closed" and in many cases has additives to prevent freezing. Such a condition requires a pump with cast iron housing that resists corrosion. An "open" loop has the possibility of associating with potable water. The actuating pump must be manufactured of stainless steel or bronze (Reference 37). A glance at one manufacturer's pump prices reveals

that the price differential for a multispeed pump is negligible (Table 5). The maximum loop head and the nature of the fluid are the main price determinants.

Table 5. Representative Pump Prices

GRUNDFOS MODEL NO.		HP	MATERIAL	WHOLESALE PRICE*	RETAIL PRICE*
UPS20-42	TWO-SPEED	1/20	CAST IRON	$51.00	$ 61.00
UM 25-18	ONE-SPEED	1/35	STAINLESS STEEL	62.00	74.00
UP26-64	ONE-SPEED	1/12	CAST IRON	64.00	76.00
UP25-42	ONE-SPEED	1/20	STAINLESS STEEL	85.00	100.00

* PRICES QUOTED AS OF AUGUST 1, 1977

Honeywell ERC is involved with the selection of hardware for a HUD residential demonstration solar air heater system (Reference 38). Prices quoted for single-speed and four-speed blower motors indicate that the cost differential is negligible. The concern in the selection of actuators for proportional control centers on long-term performance.

There are a number of major manufacturers in the valve and damper market. Barber-Colman, Honeywell, Johnson, Powers, Robertshaw and Taco are among these. Since this survey is concerned with trade-off evaluations, it seems convenient to discuss the cost differentials in terms of a single typical manufacturer who markets all types under consideration. Honeywell is such a manufacturer, whose prices and specifications are readily available for this study.

Keeping in mind that valve and damper assemblages may be utilized either to shut off, divert and mix flow, or to modulate flow as well, it is convenient to group them according to their operator characteristics rather than their functions:

- Electromechanical,
- Heat activated,
- Electronic,
- Pneumatic.

It might be noted here that in the Honeywell lines of valve and damper assemblies, the operators are often identical.

Electromechanical operators provide the traditional directional control in residential installations. The Honeywell electromechanical Modutrol motors can be spring loaded, meaning they return to their normal position when power ceases, or reversible, having two limit positions. With the appropriate control, reversible types can be positioned between fully closed and open for a modulating effect; and, depending on the model selected, limit operating times can range from several seconds to several minutes.

Honeywell also markets locally controlled heat-activated operators with longer time ranges: one temperature-sensitive bimetal damper motor, used in zone control, has a 90-degree rotation with a time range of five minutes to reach 30°; the temperature-sensitive wax valve sold by Honeywell for radiator control requires no electrical connections. Its speed of actuation is dependent on transient temperature changes with the valve usually maintaining a mid-position

relating to the temperature setpoint in the valve. Both the bi-metal and the temperature-sensitive wax operators are much cheaper than electromechanical operators. To date they appear to have found no application in solar control, but in fact the slow response of heat-activated operators may indeed render them very appropriate for proportional control in solar loops.

It is useful here to note the relative cost differentials involved when making evaluations among these categories of operators used in typical residential applications (Table 6).

Table 6. Representative Damper Operator Prices

HONEYWELL MODEL NO.	CATEGORY	DESCRIPTION	LIMIT TIME	BASE PRICE
M845 E	ELECTROMECHANICAL	SPRING RETURN 50 LB-IN. TORQUE	1 MINUTE /160°	\$196.04
M836 B	ELECTROMECHANICAL	SPRING RETURN 20 LB-IN. TORQUE	30 SECONDS / 75°	\$112.17
M833 A	HEAT-ACTIVATED	90° ROTATION 6 LB-IN. TORQUE	5 MINUTES / 30°	\$ 43.00
M754 J	ELECTRONIC	SPRING RETURN 50 LB-IN. TORQUE	50 SECONDS /160°	\$353.88

Electronic operators are more expensive than electromechanical operators, but also more versatile. They are finding increased application in complex systems as they are amenable to both local control and central control. Like electromechanical operators, electronic operators include both spring-loaded and reversible types, and also models that generate feedback signals.

As a locally controlled actuator, the electronic operator responds directly to a change in the resistance from an electronic thermostat.

It is ideal for some types of proportional control in that a rise in temperature at the thermostat opens the damper or valve accordingly. Electronic dampers are often used for local zone control. The application of locally controlled electronic valves and dampers to modulate auxiliary fuel and combustion air flow in solar systems may be particularly appropriate. Honeywell has done considerable work in evaluating the effects of excess air and flue control on seasonal furnace system efficiency for conventional systems (Reference 39). In a solar heating system, the auxiliary furnace is operated less frequently but generally at times of peak load. Evaluation of seasonal furnace efficiency in solar augmented systems is beyond the scope of this study. It requires additional software tools and points to the need for further studies.

The adaptability of electronic operators to centrally controlled microprocessor-based systems is also recognized. The Iowa State University House Solar Heat Pump System cited under Control Implementation depends upon proportional control of the collector, storage and load loops. This advanced control system utilizes three fixed-speed pumps and five electronically operated valves to modulate and direct flow as required. Three of these valves

act in combination to provide a sufficient level of heat to the house, to charge the storage tank as appropriate and to maintain a constant head through the collector loop pump. This last function is important in preventing excessive wear on the pump and reducing the possibility of its failure. Input signals to the microprocessor from the living room thermostat, the collector discharge fluid sensor and the storage tank sensor generate the output signals that determine the valve positions. Selection of the relatively expensive electronic valve operators was a consequence of the continuous tight control required in this complex four-stage heating system.

For example, under low flow conditions a 5 percent increase in flow might increase heat transfer by 25 percent. It is yet to be determined at what level a control system justifies the use of electronic over electromechanical actuating devices.

Generally pneumatic operators are eschewed in residential and small commercial systems. The limiting factor in pneumatic operation is the cost of the air compressor.

2. Large Commercial Building Systems

Proportional control is generally available, even if not applied, to all loops in a large commercial heating and cooling system, and the actuating components tend to be the valve or damper operators rather than the pumps or blowers. Throughout the industry, pneumatic components have long been accepted as the most dependable and cost-effective actuators in large systems. The basis for this lies in their relative simplicity and their amenability to local control in complex systems. To date there is limited experience with solar energy applications in large complex heating and cooling

systems. In the survey two systems representing three different approaches to collector loop control are included. All of these systems rely on state-of-the-art control as defined under control implementation.

Recently an 8000-square-foot flat plate collector system has been installed and is in operation at the Los Alamos National Security and Resources Study Center (Reference 40). In this system the solar loop operation (on/off) is centrally controlled. The HVAC system consists of pneumatic components, including the locally controlled damper and valve operators that provide zone control. Development work at Los Alamos cited under the strategy survey indicates that the system is designed to accept microprocessor-based control in the near future.

The 13,000-square-foot NASA-Langley Solar Building Heating and Cooling Test Facility has been in operation since 1976 (Reference 41). Control is provided by a Barber-Coleman pneumatic system. The solar loop is proportionally and locally controlled to maintain constant outlet temperatures within a field of rows that include seven different types of collectors. This system also was designed with the expectation that a future minicomputer or microprocessor system would be incorporated.

From these limited examples it appears that the complexity of the central control functions in large solar heating and cooling systems is leading to the design and implementation of microprocessor-based controllers. It is also apparent that the selection of pneumatic operators in a large system is totally compatible with solar energy functions. Furthermore, decisions regarding local versus central control will continue to be determined, function-by-function, within the pneumatic system. At this point it has

not become clear whether electronic actuating devices will eventually supplant pneumatic devices in large microprocessor-based control systems.

D. CONCLUSIONS

With the variety of control components and systems on the market for space heating and cooling applications, the capability certainly exists to implement any and all of the control functions and strategies that might lead to better control of solar assisted systems.

However, as functions and strategies are superimposed upon each other, conventional control systems become ever more cumbersome and complex. To the extent that the complexity gives rise to a proliferation in the number of actuators required, advanced controllers can have little impact on cost.

On the other hand, as complexity leads to more sophisticated algorithms, in particular those that depend on memory, the new developments in microcomputer technology appear increasingly attractive in the marketplace. The expanded performance and reliability capabilities inherent in this technology demand sensors and relays with compatible outputs and accuracies. Furthermore, systems-on-a-chip development encourages a greater emphasis on digital electronics reducing power consumption and enabling greater system integration.

Under Part II of this study, control system recommendations for specific applications will reflect this dichotomy in "complexity"--simplification in actuators and sophistication in control.

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

PART II. CONTROL RECOMMENDATIONS

To arrive at cost-effective control of solar assisted heating and cooling systems, the computer, the laboratory and the marketplace all must have their say. Under this part of the study, each of these voices has been heard in the evaluations of performance and control for six representative heating and cooling systems.

To investigate solar assisted climate control systems in their design state, Honeywell has available a pair of general computer programs--SUNSIM/DYNSIM--in which the components of a system are modeled mathematically and the dynamics within the system are described by a set of differential equations.

SUNSIM allows the evaluation of system performance over an entire year and is the computer tool used in most of the analyses and tradeoff studies completed in Part II of this study. DYNSIM is the tool for evaluations in which short term responses were critical.

In all of the simulations, hourly weather data from the National Climatic Center were used. During the course of the simulation work the National Climatic Center made available the SOLMET tapes, which include hourly solar data as well as meteorological data. It is noted below which tapes were used in specific regions. Where the SOLMET tapes were not available, the SUNSIM radiation model was used.

The methodology generally employed to arrive at control recommendations for each system was tightly structured:

- Select the baseline system and control;

- Simulate performance in one climate over one year;
- Define a set of control tradeoffs;
- Simulate performance under each control tradeoff and evaluate in terms of energy required and component costs;
- Select preferred control;
- Recommend a control system.

Table 7 presents a matrix of the six systems and five regions for which control recommendations are being delivered under Part II. Finally, Advanced Control Recommendations applying to all systems are also included.

Table 7. System Performance Simulations

SELECTED SYSTEM	TYPE OF NATIONAL CLIMATIC CENTER CLIMATIC DATA	SELECTED REGION	REPRESENTATIVE YEAR
A. SINGLE-FAMILY RESIDENTIAL HYRONIC SOLAR HEATING SYSTEM	HOURLY METEOROLOGICAL TDF-14 TAPE	NORTH CENTRAL REGION	MINNEAPOLIS 1957
B. SINGLE-FAMILY RESIDENTIAL ACTIVE/PASSIVE SOLAR AIR HEATING SYSTEM	HOURLY METEOROLOGICAL TDF-14 TAPE	NORTH CENTRAL REGION	MINNEAPOLIS 1957
C. SINGLE-FAMILY RESIDENTIAL HYDRONIC SOLAR HEATING/ COOLING SYSTEM	HOURLY TDF-14 AND SOLMET TAPES HOURLY SOLMET TAPE HOURLY SOLMET TAPE HOURLY SOLMET TAPE	MIDWEST REGION SOUTH REGION NORTHEAST REGION WEST REGION	OMAHA 1953 AND 1958* NASHVILLE 1955 NEW YORK 1958 ALBUQUERQUE 1962
D. SOLAR ASSISTED HEAT PUMP SYSTEM FOR A SINGLE-FAMILY RESIDENCE	HOURLY TDF-14 TAPE HOURLY TDF-14 TAPE HOURLY TDF-14 TAPE	NORTH CENTRAL REGION MIDWEST REGION WEST REGION	MADISON ST. LOUIS PHOENIX
E. COMMERCIAL BUILDING HYDRONIC SOLAR HEATING/ COOLING SYSTEM	HOURLY TDF-14 TAPE HOURLY SOLMET TAPE HOURLY SOLMET TAPE HOURLY SOLMET TAPE	MIDWEST REGION SOUTH REGION NORTHEAST REGION WEST REGION	OMAHA 1953 NASHVILLE 1955 NEW YORK 1958 ALBUQUERQUE 1962
F. COMMERCIAL BUILDING SOLAR CONCENTRATING COLLECTOR SYSTEM FOR HEATING/COOLING APPLICATION	HOURLY TDF-14 TAPE	NORTH CENTRAL REGION	MINNEAPOLIS 1956

* SOLAR PERFORMANCE DISCREPANCY BETWEEN DATA TAPES WITHIN 2.5 PERCENT.

A. SYSTEM A, SINGLE-FAMILY RESIDENTIAL HYDRONIC SOLAR HEATING SYSTEM

1. Building and System Description

In 1976 a HUD demonstration solar collector system was installed just outside of Minneapolis on a new energy-efficient house in Bloomington (Figure 4). This solar collector system was the first space heating and hot water system to use Lennox hydronic collectors. During its first months of operation, beginning in August, 1976, this system was monitored and became the data base for the validation of the software model in the Honeywell DYNSIM software package.

Figure 4. HUD Demonstration Hydronic Solar Heating System House

Inasmuch as this collector system and control were designed at the Honeywell Energy Resources Center and the software has been validated, it seemed appropriate to select this system and installation as a baseline for the control evaluations under this study.

The construction of this single-family home, now owner-occupied, results in a total envelope heat transfer coefficient of UA = 310 Btus per hour - ^{o}F:

- 1300-square-foot living area on main floor,
- 2530-square-foot total heated space,
- Triple glazed windows,
- 10-inch cellulose attic insulation (R36),
- 2-inch Dow Corning styrofoam sheathing on walls with 3-1/2-inch fiberglass batting (R25).

The fresh air infiltration rate in this tight house is estimated to be 25 cubic feet per minute.

The domestic hot water load modeled for the design and control tradeoff evaluations is based on a 1974 Rand Corporation study (Reference 42). The hourly schedule is shown in Figure 5.

Evaluations of heating loads and system performance are based on hourly weather and meteorological data for Minneapolis obtained from the National Climatic Center. Data are for the year 1957, selected as typical by the NCC.

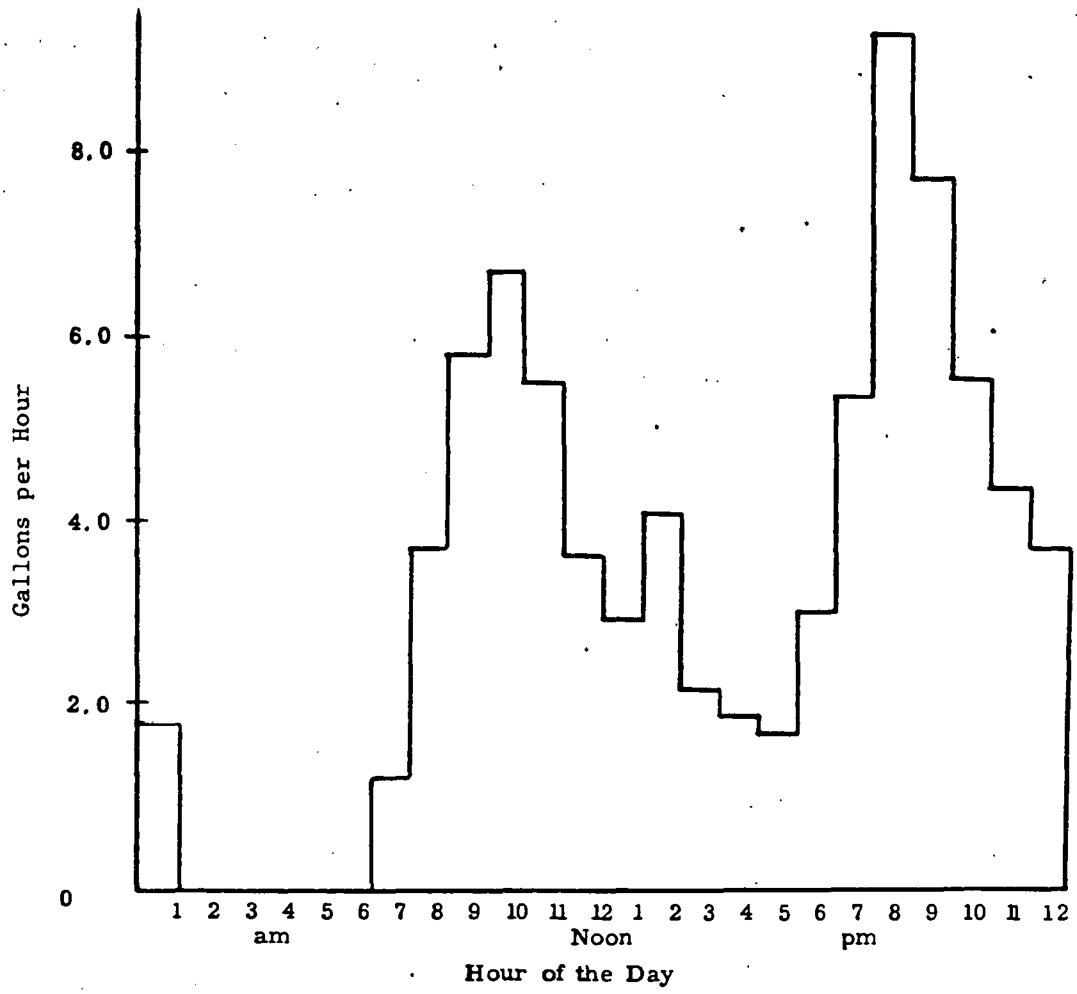

Figure 5. Single-Family Residence Hot Water Schedule

The HVAC system includes a 44,000 Btu/hr forced-air gas furnace and a large blower to provide 1050 CFM volume air flow. The conventional compressor air conditioning unit entirely controlled by the room thermostat is totally independent of the solar subsystem and therefore not a part of the simulation evaluations and control tradeoffs.

The collector subsystem includes 378 square feet of Lennox two-cover collectors--21 units in parallel--and a closed collector

loop. A 50 percent water/glycol solution is pumped through the loop at the rate of 17 gallons per minute, with a fin tube radiation coil available for purging.

Energy transfer between the closed collector loop and the closed storage loop is through a tube-and-shell heat exchanger.

The storage loop includes lengths of 1-1/4 in. copper tubing that interface with the 1000-gallon storage tank and also with the forced-air system through a water-to-air heat exchanger coil. The rate of water flow through the storage loop is 6.7 gallons per minute.

The storage tank in the basement furnace room is surrounded by walls and cover into which insulation has been blown, yielding a total envelope heat loss UA = 7 Btus/hr-^{o}F to the basement environment.

Heat transfer from the storage tank to the domestic hot water system is through a hot water preheat coil in the solar tank, connected in series with the 40-gallon DHW tank.

The control design for this baseline system includes three mutually exclusive solar modes:

- Heat house directly from solar,
- Charge the storage tank from solar,
- Heat house from the storage tank.

Figure 6a illustrates the interface of the system components (three pumps and two valves) to activate these modes. Control of the system is based on the following strategies:

- Absolute temperature control of the collector loop for the direct heating mode,
- Differential temperature control of collector/storage for charging storage,
- Absolute temperature control of the storage tank for space heating from the storage tank.

Appendix B includes a list of the actuating components and the control components in the baseline system.

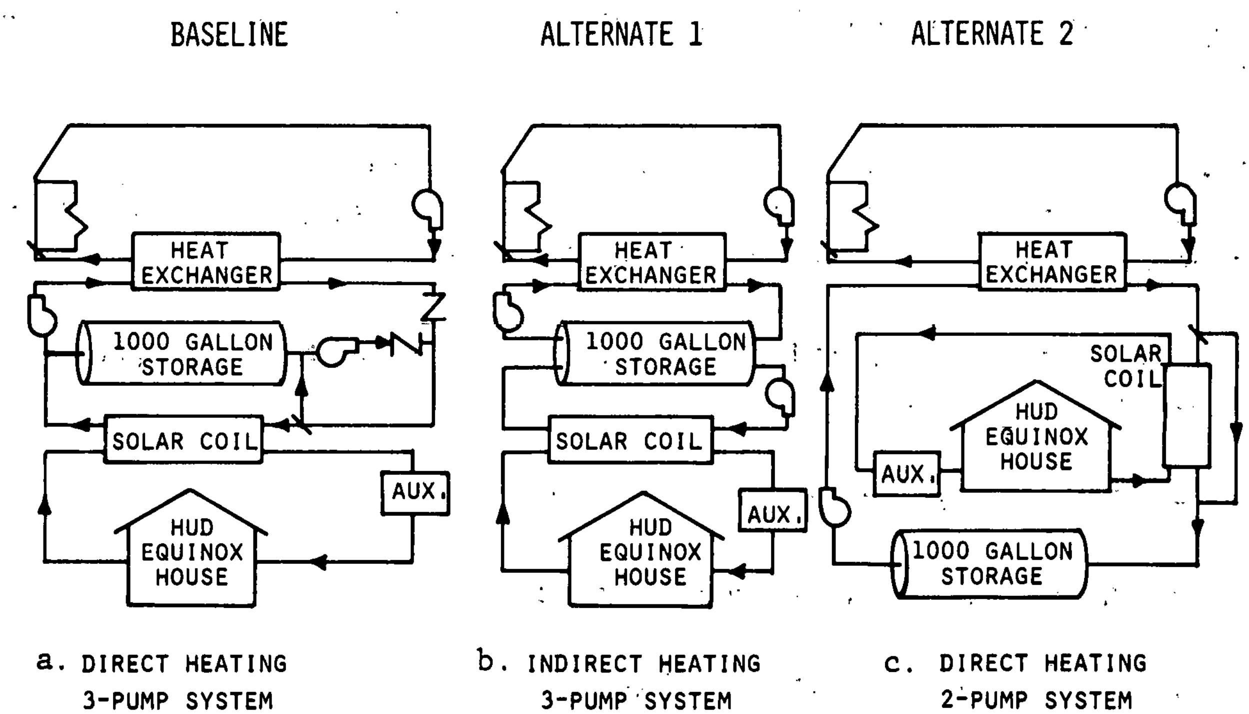

Figure 6. Alternate Design Configurations for a Single-Family Residential Hydronic Solar Collector Heating System

2. Control Tradeoff Evaluations

Tradeoff I--The initial set of system tradeoffs involves the design configuration of the solar subsystem. Two alternate designs were evaluated against the baseline configuration:

- Alternate 1: The direct solar heating mode was eliminated, along with the diverting valve in the storage loop and an absolute temperature controller (aquastat) in the collector loop. The design is shown in Figure 6b.

- Alternate 2: The system was configured as shown in Figure 6c so that the three solar modes associated with the baseline system are no longer mutually exclusive. A pump in the storage loop and the aquastat in the collector loop are both eliminated.

Each of the three systems was modeled in the SUNSIM software. Annual simulation runs were made to determine system performance. In addition to computing solar performance and auxiliary energy requirements, the hours of pump and furnace fan operation were also computed so that parasitic energy use could be determined. It turns out that this is a significant factor in evaluating energy consumption and costs.

To evaluate the energy requirements for the pumps and fan in the systems, this study has drawn on recent work connected with another Honeywell Contract--NASA Contract NAS8-32093. A system similar to this System A has been installed at William O'Brien State Park in Minnesota.

Its performance is being carefully monitored, including power consumption of all pumps and the furnace fan. Data indicate energy consumption as follows:

- Pump 1 (1/4 hp) - 335 watts,
- Pump 2 (1/6 hp) - 219 watts,
- Pump 3 (1/6 hp) - 219 watts,
- Furnace Fan (1/3 hp) - 459 watts.

These values are used in computation of parasitic energy consumption in all of the tradeoff evaluations.

Table 8 presents the annual operating costs of System A with the three design configurations. Alternate #2, the two-pump system, has been selected as the preferred design configuration. It delivers approximately the same amount of energy as the baseline system. However the initial cost saving generated by eliminating one pump and a control component is approximately $300 (1977 costs), and at 1977 energy rates for the Midwest the net annual energy savings in the system amounts to $9, using gas as the auxiliary source of heat.

Tradeoff II--Perhaps the most significant set of control tradeoff evaluations are those relating to proportional and on/off control of the collector loop.

To study both short-term system dynamics and long-term system performance, both proportional and on/off control were modeled in the DYNSIM and SUNSIM software.

Selection of appropriate parameters for the simulation was of great importance in these tradeoff evaluations. The important selections are itemized below:

Table 8. System A, Hydronic Solar Collector Heating Only System, Single-Family Residence, Minneapolis

TRADEOFF I.	SOLAR PERFORMANCE				AUXILIARY SPACE HEAT (52% EFFICIENCY)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY)		FURNACE BLOWER OPERATION		TOTAL ENERGY COSTS $/YEAR	PERCENT ENERGY SAVINGS
	SPACE HEATING BTUs/YEAR	DOMESTIC HOT WATER BTUs/YEAR	PUMPING POWER kWHs/YEAR	ELECTRICITY COST $/YEAR	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	kWHs/YEAR	ELECTRICITY COST $/YEAR		
BASELINE SYSTEM Three pumps & two valves	30.4 10^6	12.3 10^6	1359 kWHs	$54.76	35.4 10^6	$137.90 ($426.24)	8.9 10^6	$21.92 ($104.22)	1010 kWHs	$40.40	$254.98 ($625.62)	-- --
ALTERNATE #1 Three pumps & one valve	29.7 10^6	12.4 10^6	1280 kWHs	$51.20	37.1 10^6	$140.56 ($434.44)	8.8 10^6	$21.67 ($103.05)	857 kWHs	$34.28	$247.71 ($622.97)	-0.4% +0.4%
ALTERNATE #2 Two pumps & two valves	30.6 10^6	12.1 10^6	1239 kWHs	$49.56	36.2 10^6	$137.14 ($423.90)	9.1 10^6	$22.41 ($106.56)	916 kWHs	$36.64	$245.75 ($616.66)	+0.1% +1.4%

FEA 1977 natural gas rates for midwest $1.97/million BTUs

FEA 1977 electricity rates for midwest 4¢/kWH or $11.71/million BTUs

MINNEAPOLIS 1957 WEATHER

- The collector efficiency model in the software was modified to be a function of mean temperature in the collector rather than inlet temperature.

- A pumping power algorithm based on Honeywell tests of 1/20 hp Grundfos pump operation was added to the software. The algorithm reflects the linear relation between power required and level of speed control by a proportional controller as shown in the survey. For lack of published data, the assumption was made that a 1/4 hp pump in the collector loop would respond in the same linear manner (Figure 7)

- The proportional flow control algorithm used in the simulation was taken from the Rho Sigma proportional controller literature described in the survey. According to this literature, control is both proportional and linear with a temperature differential of 0 to 12°F.

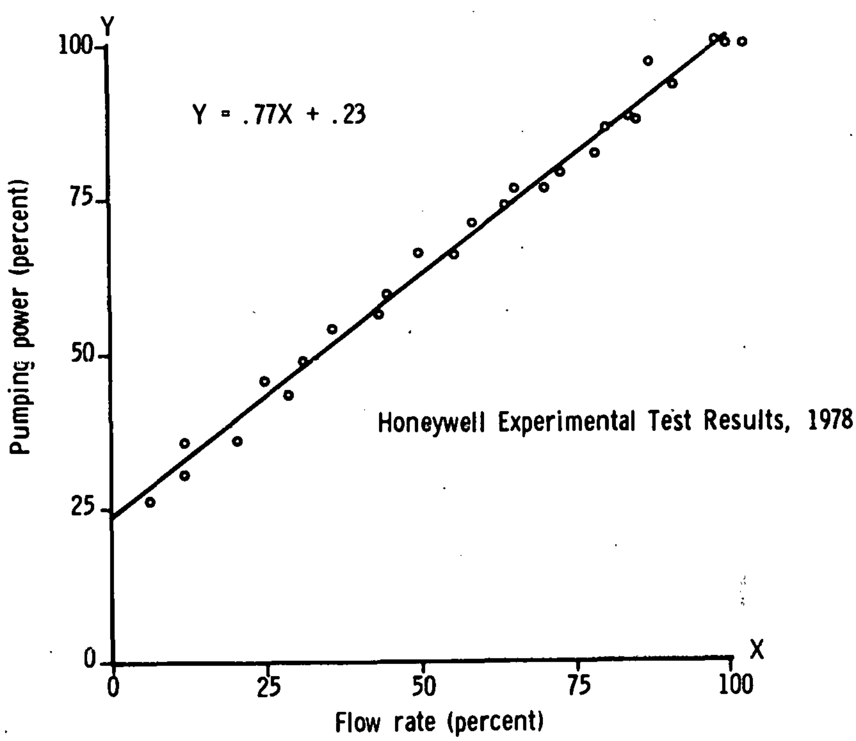

Figure 7. Variable Speed Control of a Grundfos UP25-42F 1/20 hp Pump by a Commercialized Proportional Differential Temperature Controller

To study the system response to weather-induced transients, operation was simulated over 24-hour periods using the DYNSIM software.

On a clear January day in Minneapolis, with ambient temperature between -2 and +10°F, solar energy delivered to the system under proportional control is within one-fourth of one percent as compared with energy delivered under on/off control.

On a more moderate partly cloudy day, temperatures from 20 to 30°F, proportional control delivers three-fourths of one percent more energy than on/off control. Under on/off control the collector pump cycles once when solar radiation dips significantly. The response of the system as simulated with DYNSIM is shown in Figure 8.

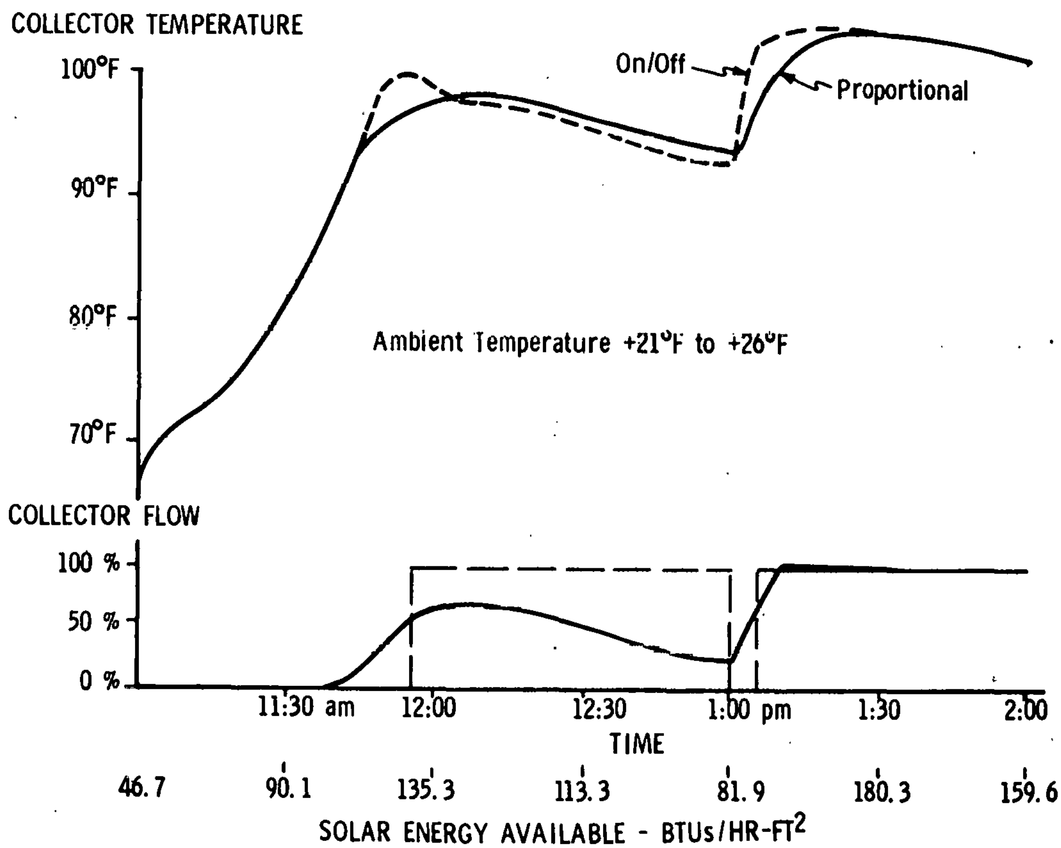

Figure 8. Collector Loop Response to a Partly Cloudy Day, Proportional and On/Off Control

To evaluate long-term performance, operation over an entire year was simulated with the SUNSIM software. The results are presented in Table 9.

The simulation results indicate that in Minneapolis the proportional control yields an additional energy savings of 2.1 million Btus natural gas, or $4.79 in one year at current rates. On the other hand, parasitic costs associated with system operation increase 89 kWHs, or $3.56 at current rates. The net annual savings with proportional control are $1.23. The cost delta involved in implementing proportional control is modest--approximately $15 at the consumer level, according to the survey.

However, uncertainties exist regarding long-term pump performance, even short-term performance for larger pumps, under other than design speed control. These uncertainties lead to the recommendation of on/off control in the collector loop.

<u>Tradeoff III</u>--The popular energy conservation strategy--thermostat night setback--was evaluated for the selected two-pump system. The survey completed as part of this control contract work cites several studies involving night setback strategies. These studies indicate that in a typical single-family residence located in the Minneapolis area (8270 degree days) 3 million Btus of natural gas can be saved with a $5^{o}F$ night setback of the thermostat. Table 10 presents the results of tradeoff simulations involving night setback. An eight-hour $5^{o}F$ night setback introduces parasitic savings as well, resulting in a net annual energy savings of $14.73 with a natural gas furnace, or $37.30 with electric resistance heat as backup.

Table 9. System A, Hydronic Solar Collector Heating Only System, Proportional versus On/Off Control of Collector Loop

TRADEOFF II.	SOLAR PERFORMANCE				AUXILIARY SPACE HEAT (52% EFFICIENCY)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY)		FURNACE BLOWER OPERATION		TOTAL ENERGY COSTS $/YEAR	PERCENT SAVINGS
	SPACE HEATING BTUs/YEAR	DOMESTIC HOT WATER BTUs/YEAR	PUMPING POWER kWHs/YEAR	ELECTRICITY COST $/YEAR	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	kWHs/YEAR	ELECTIRICTY COST $/YEAR		
ON/OFF Control Diff. Temp. 11°F, 3°F	27.1 10^6	12.4 10^6	959 kWHs	$38.36	35.4 10^6	$134.11 ($414.53)	8.8 10^6	$21.67 ($103.05)	815 kWHs	$32.60	$226.74 ($588.54)	-- --
Proportional Control Diff. Temp. 3°F, 3°F 12°F Full-On	28.5 10^6	12.5 10^6	1014 kWHs	$40.55	34.2 10^6	$129.57 ($400.48)	8.7 10^6	$21.42 ($101.88)	849 kWHs	$33.96	$225.51 ($576.88)	0.5% 2.0%

FEA 1977 Natural gas rates for midwest $1.97/MBTUs

FEA 1977 Electricity rates for midwest 4¢ kWH or $11.71/million BTUs

Minneapolis 1957 Weather Eight-hour thermostat night setback

Table 10. System A, Hydronic Solar Collector Heating Only System, Single-Family Residence, Minneapolis

TRADEOFFS III. AND IV.	SOLAR PERFORMANCE				AUXILIARY SPACE HEAT (52% EFFICIENCY)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY)		FURNACE BLOWER OPERATION		TOTAL ENERGY COSTS $/YEAR	PERCENT SAVINGS
	SPACE HEATING BTUs/YEAR	DOMESTIC HOT WATER BTUs/YEAR	PUMPING POWER kWHs/YEAR	ELECTRICITY COST $/YEAR	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	BTUs/YEAR AUX. OUTPUT	$/YEAR GAS (ELECTRIC)	KWHs/YEAR	ELECTRICITY COST		
SELECTED DESIGN Two pump system No night setback	30.6 10^6	12.1 10^6	1239 kWHs	$49.56	36.2 10^6	$137.14 ($423.90)	9.1 10^6	$22.41 ($106.56)	916 kWHs	$36.64	$245.75 ($616.66)	-- --
SELECTED DESIGN WITH NIGHT SETBACK (8 hour-5°F) Absolute temp. control of storage (90°F)	29.2 10^6	12.4 10^6	1194 kWHs	$47.75	33.7 10^6	$127.67 ($394.63)	8.8 10^6	$21.67 ($103.05	848 kWHs	$33.92	$231.02 ($579.36)	6.0% 6.0%
WITH NIGHT SETBACK Differential temp. control of storage (22°F, 6°F)	31.6 10^6	11.7 10^5	1274 kWHs	$50.96	31.2 10^6	$118.20 ($365.35)	9.5 10^6	$23.39 ($111.25)	961 kWHs	$38.44	$230.99 ($566.00)	6.0% 8.2%
WITH NIGHT SETBACK Storage reset as function of outdoor temp.	27.6 10^6	13.0 10^6	1138 kWHs	$45.52	35.3 10^6	$133.73 ($413.36)	8.2 10^6	$20.19 ($96.02)	759 kWHs	$30.36	$229.80 ($585.26)	6.5% 5.1%

FEA 1977 natural gas rates for midwest $1.97/MBTUs

FEA 1977 electricity rates for midwest 4¢ kWH or $11.72/million BTUs

MINNEAPOLIS 1957 WEATHER

It should be noted here that additional energy savings can be generated with a 10^{o}F thermostat night setback. However inasmuch as the effectiveness of this strategy is also dependent upon oversizing of the auxiliary furnace, the more conservative strategy (5^{o}F) has been selected.

The consumer cost differential between a two-stage Honeywell thermostat (T872 D1011) and the projected cost of this thermostat modified for an automatic reset capability is $62.00. Using the algorithm and economic parameters described in Appendix A, the payback period for the advanced thermostat component is computed to be under four years with a gas furnace and approximately 1-1/2 years with electric resistance heat as backup. In either case the payback is negligible when compared to the payback of the entire system.

Cited already in the survey is the impact of morning setup on peak loading of the utility, inasmuch as auxiliary heat may be called upon simultaneously by a large number of utility customers.

By modeling the capabilities of a two-stage automatic setback thermostat in the DYNSIM dynamic simulation software, this occurence was simulated. With simultaneous setup of both Stage 1 and Stage 2 at 6 a.m., an instantaneous response is demanded by the controller. Lacking this the Stage 2 call for heat from the auxiliary furnace is activated as shown in Figure 9. Several thermostat modifications were simulated including "ramped" setup and delay setup for second stage. Among these modifications, the four-hour delay almost eliminates the need for auxiliary heat on this relatively mild sunshiny autumn day (Figure 10).

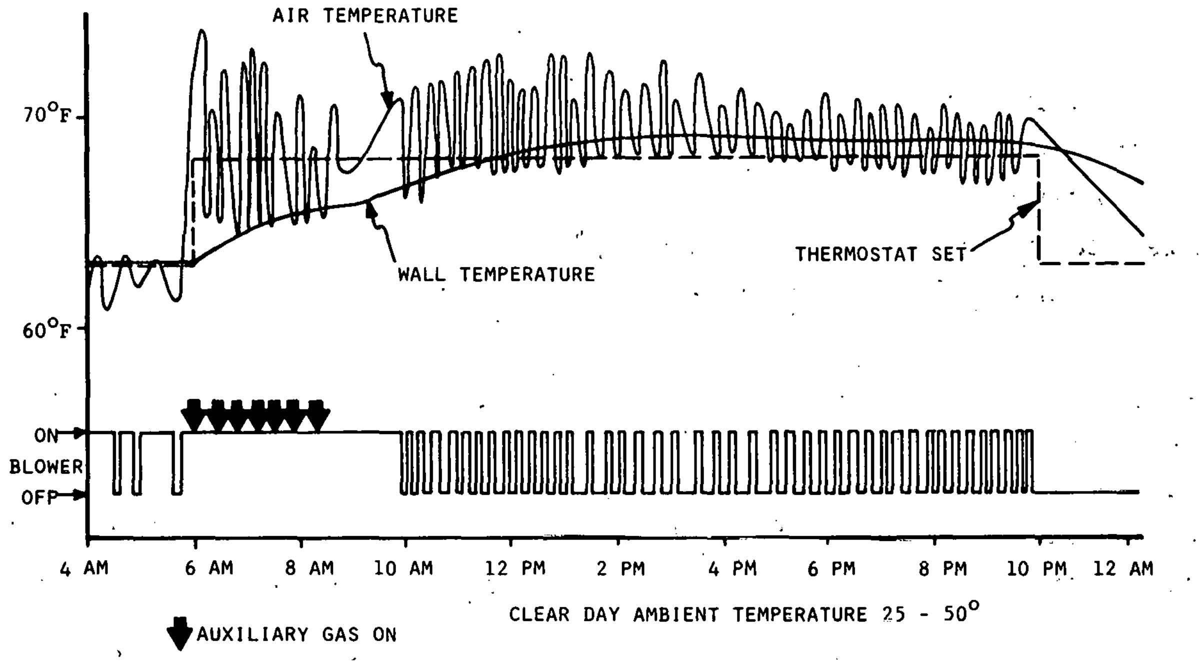

Figure 9. System Response to Thermostat Reset, Stage 1/2 Simultaneous Setup

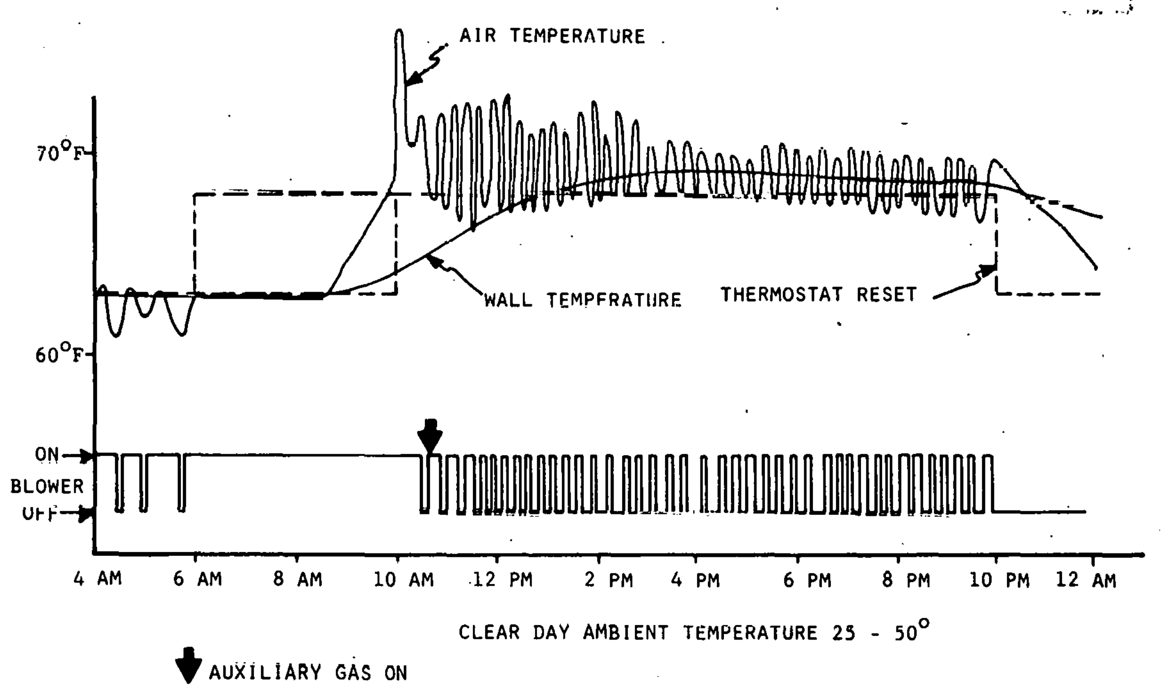

Figure 10. Stage 2 Setup Delay

The energy conservation strategy of automatic night setback is recommended as a control function for System A. But it becomes imperative that the two-stage thermostat to implement it be modified to avoid these short-term peak load occurrences without severely degrading occupant comfort levels.

Tradeoff IV--This set of control tradeoff evaluations involves control of the storage tank relative to heating the house. Three strategies have been evaluated as represented in Table 10. To evaluate the cost effectiveness of these alternate strategies, state-of-the-art Honeywell control components have been used:

STRATEGY	HONEYWELL CONTROLLER	CONSUMER COST
1. ABSOLUTE TEMPERATURE CONTROL	AQUASTAT	$ 42.00
2. DIFFERENTIAL TEMPERATURE CONTROL	DELTA T CONTROLLER AND TWO SENSORS	$140.00
3. OUTDOOR RESET CONTROL	RESET CONTROLLER	$109.00

The initial cost of implementing is $98. The energy savings generated by Strategy 2 over Strategy 1 are negligible with a gas furnace.

With electric resistance backup however, the annual energy savings are $13.36, generating an acceptable payback period for the differential temperature controller. Honeywell has a modified differen-

tial controller in the development stage that will combine the functions of the collector/storage and the storage/house controllers into a single controller. This eliminates all three controllers named above and renders Strategy 2 as most cost effective.

Tradeoff V--As the final tradeoff evaluation using the SUNSIM software, three storage tank management strategies to eliminate peak load auxiliary heating were investigated: each strategy involves charging the storage tank with electric energy during the off peak period to some predetermined setpoint.

1. A fixed setpoint, implemented by an aquastat controller and based upon coldest temperature for the location evaluated.

2. A monthly setpoint based upon average lowest daily temperature and coldest temperature by month for the location evaluated. This strategy would probably involve some kind of microprocessor implementation, or a slow-response linear controller.

3. A setpoint based on current outdoor temperature, implemented by a conventional outdoor reset controller.

The set temperatures calculated for Strategies 1 and 2 are based on the algorithms described in the survey:

- For Strategy 1, $T_{set} = 160^{o}F$;
- For Strategy 2, $T_{set} = 160^{o}F, 154^{o}F, 147^{o}F, 117^{o}F, 100^{o}F$,-,-,-, $93^{o}F, 107^{o}F, 135^{o}F, 149^{o}F$;
- For Strategy 3, $T_{set} = 155^{o}F - 1.2\ T_{amb}$.

Table 11 presents the results of these tradeoff evaluations. The monthly setpoints appear to generate the least energy costs in a single year. The cost of implementing this control approach is not available. For the time being the outdoor reset approach appears to be the best. This strategy is easily implemented with state-of-the-art components and it does indeed minimize the peak load auxiliary fuel requirements. The penalties are found in the increased blower and pumping power demand. The ratio of peak electric rates to off peak rates in the next years is still conjecture. Although peak auxiliary heating can be virtually eliminated, the substantial pumping power requirements are directly affected by the peak/off peak rate ratio selected.

3. Recommended Control System

The recommended control for System A incorporates the selections gleaned from each of the five sets of tradeoff simulations and evaluations: the two-pump system, automatic thermostat night setback, on/off control of the collector loop, differential temperature control of storage, an off peak storage strategy that is a linear function of outdoor temperature.

Details of the recommendations in terms of state-of-the-art control components, module and actuators can be found in Appendix B. The component prices are 1977 prices to the end consumer. The module cost estimate is based on the projected cost of the Honeywell Control Module designed for control of the baseline system.

Table 12 presents a summary of control component costs for the baseline system and the optimized system.

Table 11. System A, Hydronic Solar Collector Heating Only System, Off Peak Charging Two-Pump System, Minneapolis

TRADEOFF V.	SOLAR PERFORMANCE			OFFPEAK CHARGING		AUXILIARY SPACE HEAT			AUXILIARY DOMESTIC HOT WATER		FURNACE BLOWER OPERATION		TOTAL ENERGY COSTS $/YEAR	PERCENT SAVINGS
	ENERGY COLLECTED BTUs/YEAR	PUMPING POWER kWHs/YEAR	ELECTRIC $/YEAR @9.2¢/kWH	BTUs/YEAR	ELECTRIC COSTS $/YEAR	PEAK BTUs/YEAR	OFFPEAK BTUs/YEAR	ELECTRIC COSTS $/YEAR	AUX. WATER BTUs/YEAR	ELECTRIC $/YEAR @$6.73	kWHs/YEAR	ELECTRIC $/YEAR @4.6¢/kWH		
NO CHARGING Baseline system with night setback	44.4 10^6	1193.5	$109.80	--	--	10.396 10^6	21.901 10^6	$427.25	8.81 10^6	$59.29	886.8	$40.79	$637.13	--
OFFPEAK CHARGING 17:00 - 8:00														
YEARLY SETPOINT (Fixed)	35.5 10^6	1032.5	$ 94.99	49.9 10^6	$355.83	.074 10^6	.151 10^6	$ 3.01	1.94 10^6	$13.06	484.0	$22.26	$469.15	26.4%
MONTHLY SETPOINTS	37.7 10^6	1130.2	$103.98	45.6 10^6	$306.89	.073 10^6	.151 10^6	$ 2.98	3.9 10^6	$26.25	584.4	$26.88	$466.98	26.7%
OUTDOOR RESET T_{set}=155-1.2 TAMB	38.9 10^6	1216.0	$111.87	42.8 10^6	$288.04	.059 10^6	.210 10^6	$ 3.00	5.4 10^6	$36.26	710.76	$32.70	$471.87	25.9%
CHARGING 22:00-8:00 OUTDOOR RESET T_{set}=155-1.2 TAMB	39.3 10^6	1253.1	$115.29	41.4 10^6	$278.45	.071 10^6	.463 10^6	$ 5.03	5.9 10^6	$39.92	787.28	$36.21	$474.90	25.5%

Electric space heating cost: $ 6.73/million BTU - offpeak
$26.92/million BTU - peak

MINNEAPOLIS 1957 WEATHER

Table 12. Summary of Control System Costs, System A (See Appendix B for Itemized Costs)

RESIDENTIAL HYDRONIC COLLECTOR HEATING SYSTEM	CONSUMER COST BASELINE SYSTEM	CONSUMER COST OPTIMIZED SYSTEM
THERMOSTAT AND SUBBASE	$ 82.64	$ 144.46
CONTROL MODULE WITH STATE-OF-THE-ART COMPONENTS	$ 500.00	$ 484.31
WITH COMPRESSOR COOLING	ADD 20.88	ADD 20.88
ACTUATORS	$ 899.49	$ 571.84
	$1503.01	$1221.49
ADDITIONAL CONTROLS FOR OFFPEAK STORAGE CHARGING	$ 233.56	$ 233.56

The actuators contribute the bulk of the costs in the baseline control. Therefore a major goal in deriving cost effective control should be, and was in this case, the reduction in the number of actuators required. This generally leads to less complexity in the control system also.

In System A, optimized control, implemented by state-of-the-art components, can save eight million cubic feet of natural gas in one year over the baseline control (see Table 12) and can be implemented for $246 less.

B. SYSTEM B, SINGLE-FAMILY RESIDENTIAL ACTIVE/PASSIVE SOLAR AIR HEATING SYSTEM

1. Building and System Description

As the baseline system for this advanced collector design, a HUD demonstration active/passive solar heating system installed just outside of Minneapolis was selected. The residence was built in 1977 and has been occupied since January 1978 (Figure 11).

Like the HUD demonstration hydronic system, this air system, including the control system, was designed at the Honeywell Energy Resources Center.

This single-family house meets all requirements of the Uniform Building Code, the Minnesota State Building Code and the Minnesota State Energy Code. The total overall envelope heat transfer coefficient for the building is 400 Btus per hour- ^{o}F. The living space on two floors totals 2100 square feet.

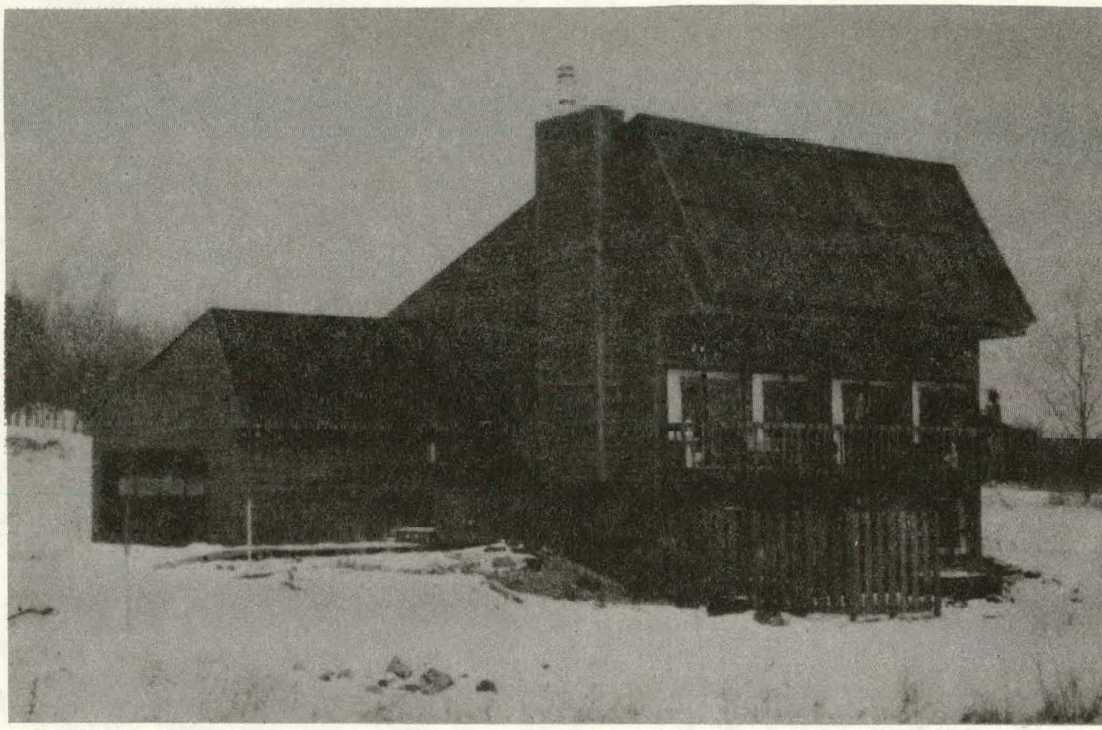

Figure 11. HUD Demonstration Active/Passive Solar Heating System House

The passive solar capability includes a number of features worth noting:

- Triple glazed windows on all sides;
- Wall on the south side of house contains 218 square feet of triple glazed window for passive solar collection;
- Overhang on window wall to prevent excessive solar heat gain in summer;
- Automatic passive air circulation control.

The domestic hot water load modeled for the design and control tradeoff evaluations is that modeled for the System A evaluations (Figure 5).

Evaluations of heating and cooling loads as well as system performance are based on hourly weather and meteorological data for Minneapolis obtained from the National Climatic Center. The selected year is 1957.

The active solar assisted heating system, controlled by a two-stage thermostat with automatic heating/cooling changeover, includes the following components:

- Twenty-two Lennox air collectors, two-cover, mounted two in series;
- Lennox air handler with a 4 speed, 1/2 hp, 800 cfm fan;
- Well insulated bin containing 400 cubic feet of rock storage medium;
- Economizer cooling with automatic control of outside dampers;
- A natural gas furnace with 50,000 Btu per hour output;
- Compressor air conditioning.

Figure 12 presents a schematic of the entire system.

Solar heating, either directly from the collectors or from storage, is activated with the first-stage call for heat from the thermostat. If neither source is available, the first-stage call will activate the auxiliary gas furnace. The second-stage call for heat will always activate the auxiliary gas furnace.

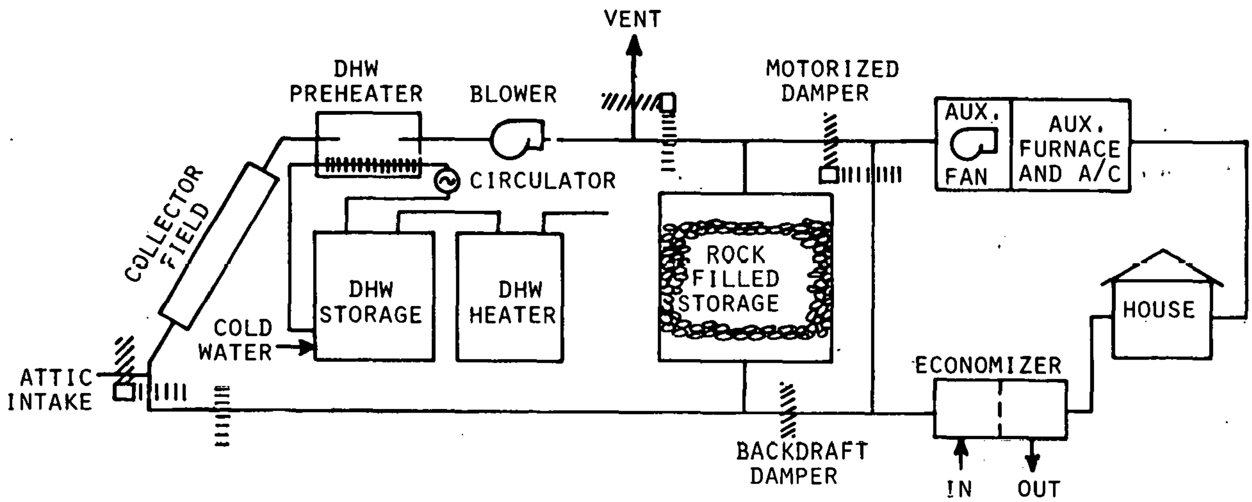

Figure 12. Single-Family Residential Air Collector Heating System, Baseline Design

Collector loop control is based on a collector/storage temperature differential. The storage bin relies on absolute temperature control.

For summer comfort, a manual switch uncouples the storage mass from the control loop. Thus energy is transferred from the collectors only when there is a call for heat. Otherwise the collector dampers are in a venting position.

The solar collectors also provide energy to the domestic hot water preheater through a water coil in the duct. A pump is activated along with the solar blower when there is a significant temperature differential between the solar collector sensor and the sensor in the preheater tank.

If the enthalpy sensor indicates that the outdoor air is satisfactory for cooling, a call for first-stage cooling from the room thermostat activates the economizer dampers and the furnace fan. A call for second-stage cooling activates the compressor air conditioning unit. Over-temperature in the collectors is prevented by two sets of venting dampers open to the outside.

Appendix B includes a list of the actuating components and the control components in the baseline air collector system.

2. Tradeoff Evaluations

For these evaluations, the following parasitic power demands have been assumed:

Air Handler Fan--
- 172 watts when flow is directly to the load,
- 195 watts when flow is through the storage bin,
- 30 watts for low-speed purge.

Furnace Blower--
- 439 watts.

Tradeoff I--The effect of night thermostat setback was evaluated in the first set of tradeoff simulations. As expected, an eight-hour 5°F setback implemented by a two-stage thermostat with automatic reset capabilities is cost effective.

Table 13 presents the detailed results of the simulations. For a system with seasonal furnace efficiency of 52 percent, more than 8 MCF of natural gas can be saved over one year. There are also small parasitic savings yielding total savings of $17.60 according to current fuel rates, as itemized in Appendix A.

The projected consumer cost differential for a two-stage thermostat modified to include automatic reset is $62, resulting in a payback period just over three years. With electricity as the backup energy source this payback period is substantially less. In either case the payback for this improved control device renders it cost effective within the system design.

Tradeoff II--It is certain that in areas where electricity is the backup energy source, storage management strategies must be developed to relieve peak load usage of the electric power. For this system three tradeoff strategies for off peak storage charging were evaluated:

- Fixed minimum storage temperature limit for the entire season based upon the coldest temperature for that location;
- Monthly minimum storage limits based on coldest monthly averages;
- Variable storage limit based on current ambient temperature.

The algorithms to compute these minimum storage temperatures and the implementing control devices are the same as those for System A.

Table 13. Active/Passive Solar Air Collector Heating System Control Tradeoffs, Minneapolis

GAS COST COMPUTATIONS INCLUDE A 52% SEASONAL FURNACE EFFICIENCY AND AN 80% EFFICIENT WATER HEATER	SOLAR PERFORMANCE					AUXILIARY SPACE HEATING		AUXILIARY DOMESTIC HOT WATER		FURNACE BLOWER OPERATING POWER		TOTAL ENERGY COSTS	
	PASSIVE SPACE HEATING BTUs/YR.	ACTIVE SPACE HEATING BTUs/YR.	AIR HANDLER POWER kWH/YR.	DOMESTIC HOT WATER BTUs/YR.	ELECTRIC $/YR. @ 4¢/kWH	BTUs/YR.	$/YR. GAS @ $2/MBTU	BTUs/YR.	$/YR. GAS @ $2/MBTU	kWHs/YR. @ 439W/HR.	$/YR. @ 4¢/kWH	$/YEAR	PERCENT SAVINGS
BASELINE SYSTEM-NO NIGHT SETBACK-GAS AUXILIARY	19.7 10^6	28.8 10^6	278	10.9 10^6	$11.12	33.8 10^6	$130.00	10.4 10^6	$26.00	967	$38.68	$205.80	--
WITH NIGHT SETBACK 8 HOURS, 5°F	19.7 10^6	28.3 10^6	276	10.9 10^6	$11.04	29.4 10^6	$113.08	10.4 10^6	$26.00	952	$38.08	$188.20	8.6%
OFF-PEAK STRATEGIES:4:1 RATIO DAYTIME/NIGHTTIME RATES ELECTRIC AUXILIARY					ELECTRIC $/YR. @9.2¢/kWH	BTUs/YR. DAY-NIGHT	$/YR.ELEC. @$26.92-$6.73/MBTU	BTUs/YR. NIGHT	$/YR.ELEC. @$6.73/MBTU	kWHs/YR. DAY-NIGHT	$/YR. @4.6¢/kWH		
NO STRATEGY	19.7 10^6	28.3 10^6	276	10.9 10^6	$25.39	5.6 + 23.8 10^6	$298.81	10.4 10^6	$69.99	952	$43.79	$437.98	--
CHARGE STORAGE TO 119°F	19.7 10^6	17.4 10^6	237	12.8 10^6	$21.80	.3 + 39.9 10^6	$276.60	8.5 10^6	$57.21	407	$18.73	$374.34	14.5%
CHARGE STORAGE-RESET CONTROLLER													
SINGLE BIN	19.7 10^6	24.0 10^6	257	12.0 10^6	$23.64	.3 + 33.4 10^6	$232.86	9.3 10^6	$62.59	543	$24.98	$344.07	21.4%
DIVIDED BIN	19.7 10^6	25.6 10^6	268	11.4 10^6	$24.66	.3 + 31.8 10^6	$222.09	9.9 10^6	$66.63	478	$21.99	$335.37	23.4%
DOUBLE DIVIDED BIN	19.7 10^6	27.9 10^6	274	10.9 10^6	$25.21	.3 + 29.5 10^6	$206.61	10.4 10^6	$66.69	620	$28.52	$327.03	25.3%
CHARGE STORAGE-MONTHLY RESET													
SINGLE BIN	19.7 10^6	19.7 10^6	246	12.4 10^6	$22.63	.3 + 37.7 10^6	$261.80	8.9 10^6	$59.90	451	$20.76	$365.09	16.6%
DIVIDED BIN	19.7 10^6	24.7 10^6	267	11.5 10^6	$24.56	.3 + 32.6 10^6	$227.47	9.8 10^6	$65.95	421	$19.38	$337.36	23.0%
DOUBLE DIVIDED BIN	19.7 10^6	27.4 10^6	271	11.1 10^6	$24.93	.3 + 30.3 10^6	$212.00	10.2 10^6	$68.65	503	$23.13	$328.71	24.9%

For System B, the computed temperature limits are:

- Strategy 1, $T_{set} = 119^{\circ}F$;
- Strategy 2, $T_{set} = 121^{\circ}F$, $118^{\circ}F$, $115^{\circ}F$, $97^{\circ}F$, $86^{\circ}F$, -,-,-, $82^{\circ}F$, $90^{\circ}F$, $108^{\circ}F$, $115^{\circ}F$;
- Strategy 3, $T_{set} = 118^{\circ}F - 2/3\ T_{amb}$.

As in System A, the peak period heating load is virtually eliminated under all strategies. System B parasitic power requirements are more modest than those in System A. The outdoor reset controller introduces a cost differential of \$99. However the electricity savings (4:1 peak/off peak difference) are \$93 in the first year at an off peak rate of 2.3¢/kWH. Clearly, in System B reset control is the best among those strategies evaluated and presented in Table 13.

Tradeoff III--To determine whether the storage bin might be better configured to handle off peak strategies, one set of tradeoff evaluations modified the bin itself. By dividing the bin horizontally into two equal portions, with the appropriate dampers and control, the upper portion can be used for off peak storage and the lower for solar storage. Simulations were performed with this configuration as well as with a double-sized bin in the same configuration.

As detailed in Table 13, close to \$10 can be saved by dividing the storage bin and incorporating additional dampers, motor and damper control into the system. In view of the quest for actuator simpli-

city that underlies this study, this increase in complexity was rejected as not being cost effective.

3. Recommended Control System

The recommended control system includes automatic thermostat night setback, economizer control, and a single storage bin for implementing the off peak storage charging strategy.

The system configuration recommended is somewhat simpler than the baseline design that has already been built.

Experimental work completed by Honeywell ERC indicates that the Lennox air collectors hold up well in a stagnation environment, eliminating the need for collector vents. Therefore one damper assembly (motor plus two dampers) has been removed from the system to provide for a bypass mode rather than a vent mode. The alternate design is shown in Figure 13.

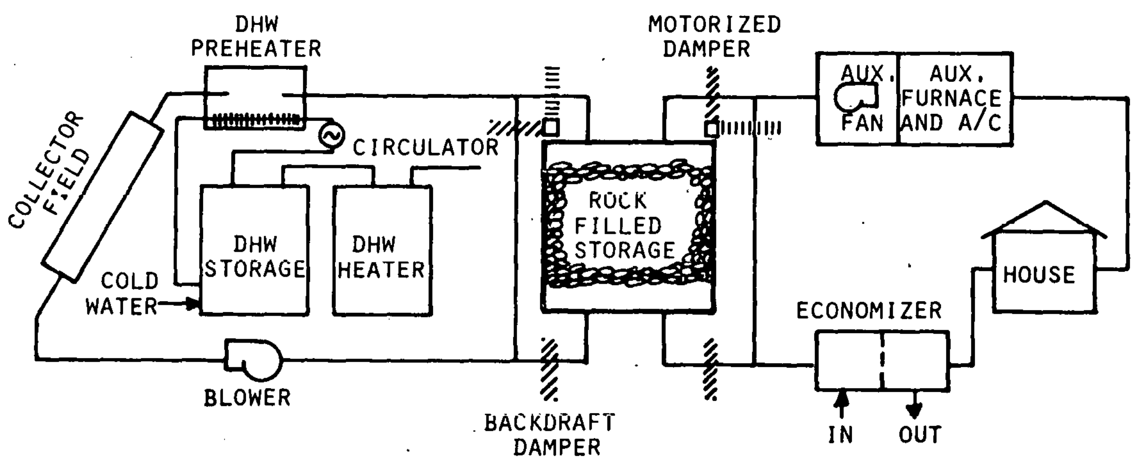

Figure 13. Single-Family Residential Air Collector Heating System, Recommended Design

Table 14 presents a summary of the baseline and the recommended control systems, as implemented with Honeywell state-of-the-art components. Detailed lists of the components can be found in Appendix B.

Cost savings in the implementation of the optimized system amount to $858. The first year energy savings with the improved control have been documented above. The principal savings accrue from the reduced number of actuating components and the resulting simplification of control implementation.

Table 14. Summary of Control System Costs, System B (See Appendix B for Itemized Costs)

RESIDENTIAL AIR COLLECTOR HEATING SYSTEM	CONSUMER COST BASELINE SYSTEM	CONSUMER COST OPTIMIZED SYSTEM
THERMOSTAT AND SUBBASE	$ 82.64	$ 144.46
CONTROL MODULE WITH STATE-OF-THE-ART COMPONENTS	$ 674.37	$ 455.80
WITH COMPRESSOR COOLING	ADD 20.88	ADD 20.88
WITH ECONOMIZER	ADD 69.63	ADD 69.63
DOMESTIC HOT WATER	ADD 112.00	ADD 112.00
ACTUATORS	$1250.98	$ 549.72
WITH ECONOMIZER	ADD 284.47	ADD 284.47
	$2494.97	$1636.96
ADDITIONAL CONTROLS FOR OFFPEAK STORAGE CHARGING	ADD $ 233.56	ADD $ 233.56

C. SYSTEM C, SINGLE-FAMILY RESIDENTIAL HYDRONIC SOLAR HEATING/ COOLING SYSTEM

1. Building and System Description

For purposes of these control evaluations a typical new house with 1500 square feet of floor space has been selected. This selection is based on work recently completed at the Honeywell Energy Resources Center under an ERDA contract (Reference 43).

The new home construction characteristics are outlined by region in Table 15. The overall envelope heat transfer coefficients become:

- 480 Btus/hr-^{O}F for the North Central region,
- 454 Btus/hr-^{O}F for the Northeast,
- 444 Btus/hr-^{O}F for the West,
- 467 Btus/hr-^{O}F for the South.

Heat transfer due to air infiltration is assumed to be 10 percent of the envelope heat transfer in each case. The solar load on the windows is also modeled, using window data from Table 15.

The domestic hot water load is taken from a 1974 Rand Corporation study (Reference 42). The hourly schedule is shown in Figure 5. The solar heating/cooling system model for this single-family residence reflects previous design work by the Energy Resources Center.

Table 15. Single-Family (New) Construction Characteristics/Resistance

Surface	Region	Construction Description	R-Value
Exterior Wall	NE, NC	Wood siding	0.81
		1/2" fiberboard	1.32
		3 1/2" insulation	11.
		2x4 studs	4.35
		1/2" gypsum board	0.45
	S	Brick veneer	0.44
		1/2" fiberboard	1.32
		3 1/2" insulation	11.
		2x4 studs	4.35
		1/2" gypsum board	0.45
	W	Stucco 5/8"	0.13
		3 1/2" insulation	11.
		2x4 studs	4.35
		1/2" gypsum board	0.45
Ceiling		Insulation	
	NE	6" Batts	19
	NC	6" loose fill	13
	S	6" loose fill	13
	W	6" Batts	19
		2x6 rafters	6.85
		1/2" gypsum board	.45
Windows*	NE, NC	Single pane with storms	1.79
	S, W	Single pane without storms	.88
Sliding Glass Door*	NE, NC S, W	Double pane, 1/4" air space(Metal Frame)	1.40
Door*	NC	1 1/2" solid wood door, with Wood/glass storm door	3.70
	NE, S, W	1 1/2" solid wood door	2.04
Walls Below Grade	NE, NC	12" concrete blocks including earth	14.29
		3/4" styrofoam and furring	3.00
		1/2" gypsum board	.45
	S, W	No Basement	
Basement/ Bottom Floor	NE, NC	Carpet and pad	2.08
	S, W	4" concrete	.32

*R values include surface resistances

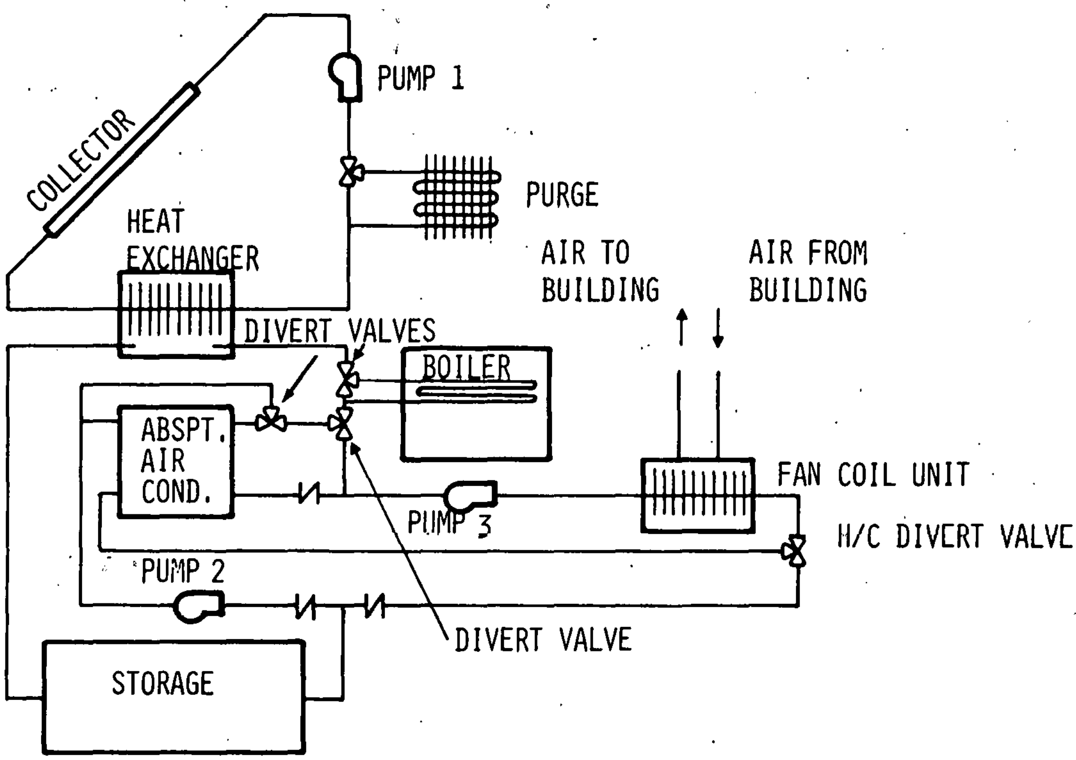

Figure 14. Hydronic Solar Collector Heating/Cooling System for Single-Family Residence

The solar subsystem portion closely resembles the selected two-pump heating system, with the addition of economizer control, a three-ton ARKLA absorption cooling unit (Figure 14) and the associated pumps and valves. The collector field and the storage tank are sized as they were for the heating-only system--378-square-foot gross collector area and a 1000-gallon storage tank. Other

design parameters also remain the same: two-cover Lennox collectors, a 50 percent glycol solution in the collector loop and a collector loop flow of 15.2 gallons per minute.

The storage loop flow is set at 11.0 gallons per minute for the entire year, based on the ARKLA absorption cooling unit requirements. A third pump is added to the system for delivery of the hot or chilled water, at 7.2 gallons per minute.

The H/C valve and the divert valve are controlled from the thermostat. For the purposes of these studies, Pump 1 is defined as a 1/4 hp single-speed pump; Pumps 2 and 3 are defined as 1/6 hp single-speed pumps. The heat exchanger in the solar loop has an effectiveness of 0.6.

The absorption cooling unit modeled is the ARKLA Model WF36 three-ton unit with the following specifications;

- Design hot water flow, 11 gpm,
- Design chilled water flow, 7.2 gpm,
- Design hot water inlet temperature, 195^{o}F,
- Design chilled water outlet temperature, 45^{o}F,
- Design delivered capacity, 36,000 Btu/hour,
- Time constant, 6 minutes.

The fan coil modeled is the ARKLA vertical fan coil filter assembly, Model VFCF 36-90, with the following specifications:

- Total heating capacity, 90,000 Btus/hour,
- Rated hot water temperature, 175°F,
- Total cooling capacity, 36,000 Btus/hour,
- Rated chilled water temperature, 45°F,
- Rated air flow, 1200 cfm,
- Rated water flow, 7.2 gpm,
- Effectiveness at design conditions, 0.5.

Control options for this residential system will be evaluated in the four cities representative of four regions in the continental United States (Table 16).

Table 16. Heating and Cooling Regions

REGION	WEIGHTED AVERAGE ANNUAL HEATING DEGREE DAYS	LOCATION SELECTED					
		CITY	DEGREE DAYS	WINTER DESIGN (DB)	SUMMER DESIGN (DB/WB)	LATITUDE	LONGITUDE
MIDWEST	6,345	OMAHA	6,612	-12°F	94°/78°F	41.5°	96.0°
NORTHEAST	5,470	NEW YORK	5,219	+ 6°F	91°/76°F	40.8°	74.0°
SOUTH	2,795	ATLANTA (NASHVILLE DATA)	2,961	+14°F	92°/77°F	33.8°	84.5°
WEST	3,515	ALBUQUERQUE	4,348	+ 6°F	94°/65°F	35.0°	106.6°
NORTH CENTRAL		MINNEAPOLIS	8,158	-19°F	89°/75°F	44.7°	93.0°

2. Tradeoff Evaluations

Six distinct sets of tradeoff evaluations were completed for this heating/cooling system in each of four regions in the continental United States. The representive locations are the first four described in Table 16.

The tradeoff studies address the following strategies sequentially:

I. Thermostat night setback;

II. Effect of thermostat anticipation on absorption cooling performance;

III. Proportional control of the absorption unit;

IV. Need for an auxiliary boiler;

V. Differential temperature control of storage;

VI. Heating season off peak strategies.

For all regions, the seasonal gas furnace system efficiency has been estimated at 60 percent. Itemized results of the simulation and energy cost evaluations can be found in Appendix C for each representative city. Tables C-1 through C-9 present summaries of these results.

Tradeoff I--The initial set of tradeoff simulations evaluated eight-hour night setback of 5^{o}F. Results for the four representative cities are presented in Table 17.

Table 17. Simulation Results, 5°F Night Setback

REGION	REPRESENTATIVE CITY	REFERENCE	COST SAVINGS IN $/YEAR INCLUDES PARASITIC COSTS	PERCENT SAVINGS
NORTH CENTRAL	OMAHA	TABLE C-1	$14.87 ($50.47)	6.7% 7.4%
NORTHEAST	NEW YORK	TABLE C-7	$27.99 ($73.69)	7.6% 7.7%
WEST	ALBUQUERQUE	TABLES C-5, C-6	$10.61 ($25.34)	8.2% 10.2%
SOUTH	ATLANTA	TABLES C-3, C-4	$13.06 ($34.55)	6.5% 7.9%

AUXILIARY FUEL IS NATURAL GAS (ELECTRICITY IN PARENTHESES).
FEA 1977 RATES BY REGION ARE LISTED IN APPENDIX A.

Tradeoff II--The most significant control tradeoffs in this system relate to the operation of the absorption cooling unit. A three-ton residential unit typically requires a "warmup" time of 15 to 20 minutes. Recent studies have been directed toward the reduction of on/off cycling frequency by diverting the chilled water to storage when the building load is small relative to the cooling capacity of the chiller (References 44 and 45). This, of course, requires the addition of a cold-side storage tank to the residential heating/cooling system, or the elimination of hot-side storage during the cooling season.

An alternate approach to this cycling problem is through the thermostat anticipator. (For a thorough analysis of the anticipator/deadband/cycling interface, see Reference 46).

In a cooling mode, the level of feedback temperature rise in the thermostat controls the cycling of the chiller. This level of anticipation is generally based on the comfort requirements of the building occupants. To the extent that a solar-fired absorption unit operates at off design conditions, it may be possible to minimize anticipation without severely affecting occupant comfort.

This set of tradeoff evaluations examines this thermostat modification and its impact on the cooling system performance.

The ARKLA Solaire 36 unit (three-ton) was modeled and performance simulated at a variety of load levels with and without an anticipator in the thermostat.

The model included a time constant of six minutes corresponding to a "warmup" time of 20 minutes for the ARKLA unit. The results are significant:

- At design conditions--195°F inlet temperature to the generator and 85°F inlet condensing water temperature--a steady state COP of 0.72 exists, according to the ARKLA specifications.

- However, a typical thermostat with anticipation generates several cycles within a single hour, which tends to reduce the average COP. For instance, as shown in Figure 15, at a light 10,000 Btu/hour cooling load the interface of the ARKLA dynamics with the thermostat dynamics results in a cooling cycle of 20 minutes and an average COP of 0.42.

- When the anticipator is disabled, the thermal mass of the building becomes the driving constant. For the single-family residence under study here, with a thermal capacitance of 13,000 Btus/°F, the cooling cycle is extended to 170 minutes with an average COP of 0.64.

- These differences in absorption performance, shown for a partial load condition in Figure 15, are more dramatic at lighter loads and, of course, disappear when the system approaches full load.

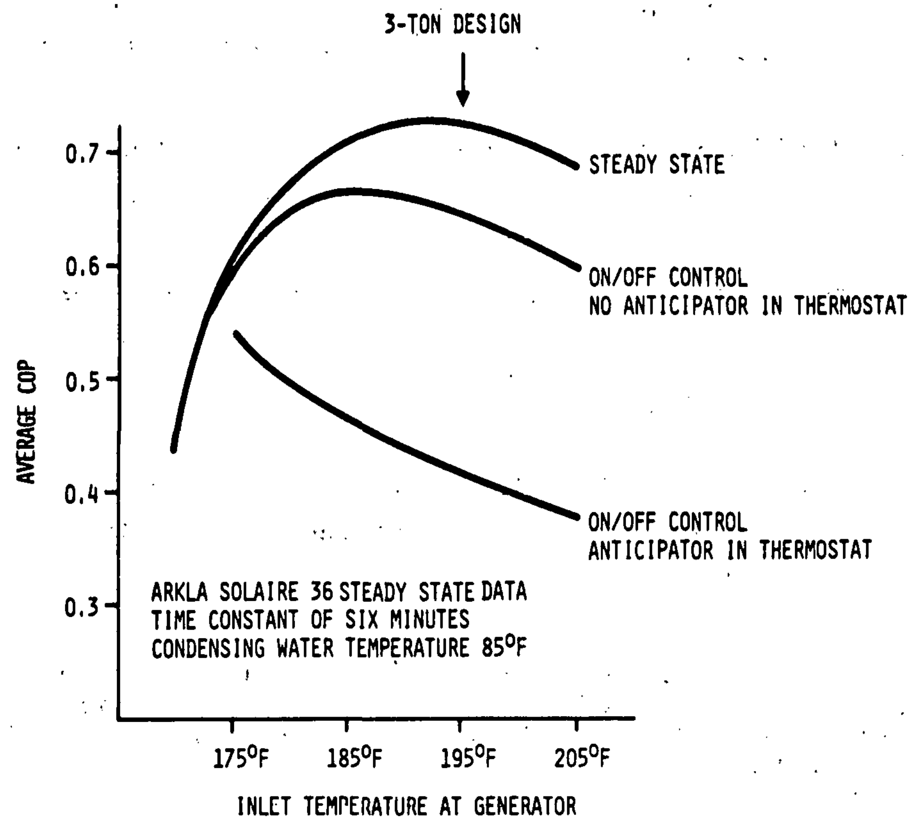

Figure 15. Three-Ton Absorption Chiller Response to Part Load (10,000 Btus/Hour) Control Tradeoffs

Tradeoff III--As cited previously, current experimental work with solar absorption cooling systems addresses the cycling problems associated with on/off control. On the other hand, attempts are being made to eliminate cycling of the ARKLA unit by controlling the inlet flow to the generator, thereby driving down the capacity to meet the building load. Control can be implemented by a modulating valve so that the various pumps continue to operate at design

speed. With proportional control, the steady state COP of the unit is maintained as long as there is a cooling load, as suggested in Figure 15.

This proportional mode of operation was modeled in the SUNSIM software. Operation of the system in Omaha was simulated over one year and results tabulated as shown in Tables C-1 and C-2. Indeed the average season COP was improved, from 0.68 to 0.74; however in the application, the penalties for this mode of operation show themselves in the parasitic power requirements that yield an additional net annual energy cost of \$25-\$27, depending on the choice of backup energy (Table C-2). For the purposes of this study, on/off control, with minimum anticipation, is the selected mode of control for the absorption unit.

However as more experimental information on the ARKLA unit becomes available, further studies on this question should be undertaken. At such time the question of minimum anticipation in the thermostat vis-a-vis occupant comfort can also be addressed.

<u>Tradeoff IV</u>--The ever present issue in system design is the hardware question. The performance and economic evaluations related to Systems A and B (residential heating only) clearly show that the actuator costs overpower the control cost considerations. Therefore the heating/cooling system evaluations are directed toward the auxiliary boiler that boosts the temperature of the fluid entering the absorption generator.

In Table 18 the contribution of the auxiliary boiler in the four regions is noted. In the Northeast (New York) it is apparent that this residential heating/cooling system may not be appropriate at all. The total number of cooling hours during a typical year is 118, with a thermostat setting of 78°F.

With reference to Albuquerque, it is clear from Table 18 that an auxiliary boiler is not required for a solar-fired absorption cooling system.

Of interest in this set of tradeoff evaluations is the contribution of the boiler to cooling operation in the North Central Region (Omaha) and in the South (Atlanta).

Figure 16 presents a profile of the annual 1.1 million Btu cooling left unsatisfied when the boiler is removed from the system in Omaha. On the basis of the profile, together with the total load and hours not satisfied, the recommended system for Omaha eschews an auxiliary boiler.

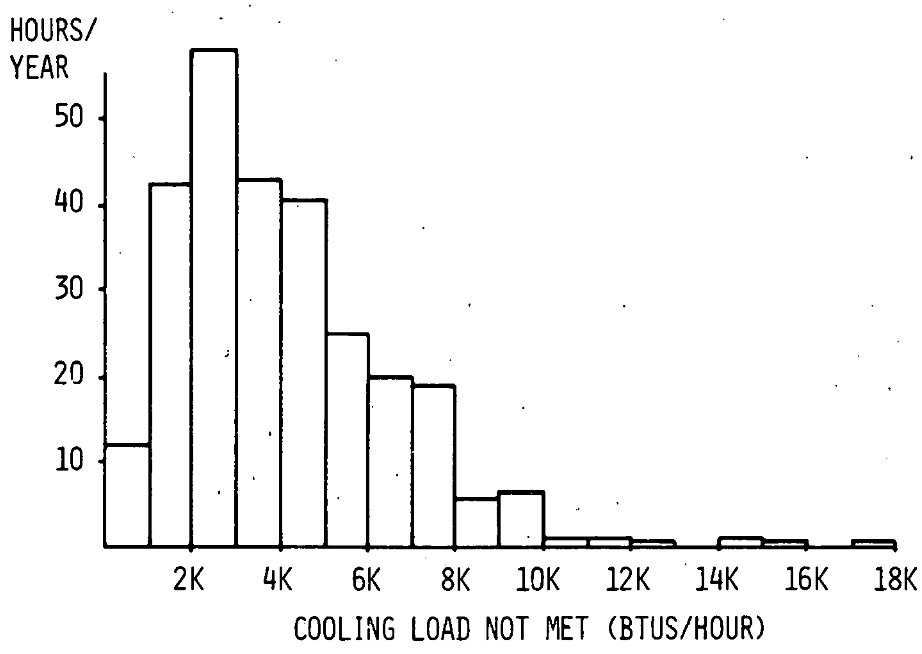

Figure 16. Residential Absorption Cooling Performance in Omaha Without Auxiliary

Table 18. System C, Effect of Auxiliary Cooling (Boiler) on System Performance and Occupant Comfort

REGION	REPRESENTATIVE CITY	NET COOLING LOAD TO BE SATISFIED BY ABSORPTION COOLING BTUS/YEAR	MINIMUM AUXILIARY ENERGY INPUT TO ABSORPTION UNIT TO SATISFY LOAD (BTUS/YEAR)	NO AUXILIARY: COOLING LOAD NOT SATISFIED	ANNUAL ENERGY COST SAVINGS WITHOUT AUXILIARY COOLING
MIDWEST	OMAHA	11.5×10^6 /502 HOURS	2.5×10^6	1.1×10^6 BTUS /135 HOURS	$18.11 ($40.92)
NORTHEAST	NEW YORK	3.1×10^6 /118 HOURS	0.3×10^6	NOT APPLICABLE	NOT APPLICABLE
WEST	ALBUQUERQUE	9.8×10^6 /299 HOURS	0.01×10^6	1,300 BTUS /1 HOUR	$ .02 ($ 0.08)
SOUTH	ATLANTA	13.2×10^6 /725 HOURS	5.5×10^6	3.4×10^6 BTUS /548 HOURS	$24.23 ($64.59)

AUXILIARY FUEL IS NATURAL GAS (ELECTRICITY IN PARENTHESES).
FEA 1977 RATES BY REGION ARE LISTED IN APPENDIX A.

On the other hand, in Atlanta the load left unsatisfied by removal of the boiler is 3.4 million Btus in a typical year. The profile of this load (Figure 17) indicates the need for an auxiliary boiler and the associated control components.

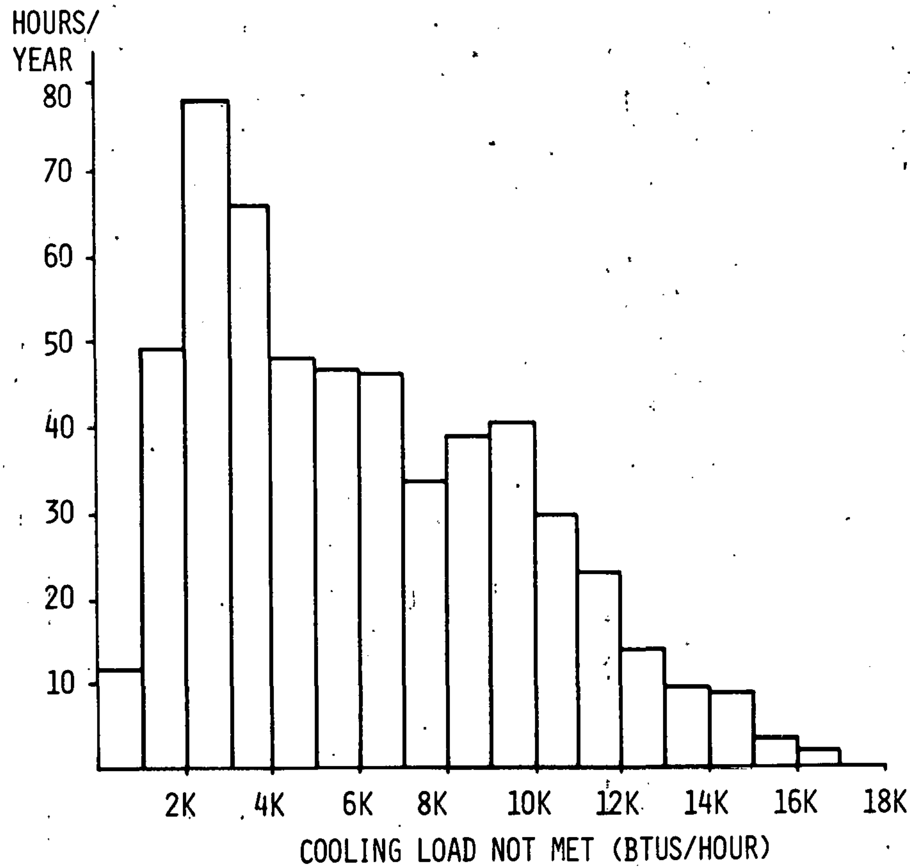

Figure 17. Residential Solar Absorption Cooling Performance in Atlanta Without Auxiliary

Once the auxiliary boiler is available, additional control innovations to meet the load requirements can be considered. From Figure 18 it is evident that conditions exist under which the minimum inlet temperature (170°F) is not sufficient to satisfy the cooling

load. Performance of the Atlanta system was evaluated under two different control strategies:

(1) When there is a call for cooling, the generator inlet fluid temperature is boosted to the minimum 170°F.

(2) When there is a call for cooling, the minimum generator inlet fluid temperature is a function of ambient temperature: $T_{inlet} = 170°F + 1/2\ (T_{amb} - 78°F)$.

Strategy 1 requires auxiliary energy of 5.5×10^6 Btus over one year and leaves one percent of the cooling load unsatisfied.

Strategy 2 leaves the auxiliary energy contribution virtually unchanged and does satisfy the entire load over a cooling season.

The cost differential associated with Strategy 2 is not significant with conventional control implementation:

Strategy 1:	Aquastat at	$ 42
Strategy 2:	Reset Controller at	$112
		+ $ 70

However the far more significant cost of the auxiliary boiler really demands that it be effectively controlled to satisfy load requirements. Therefore Strategy 2 is selected for the Atlanta system.

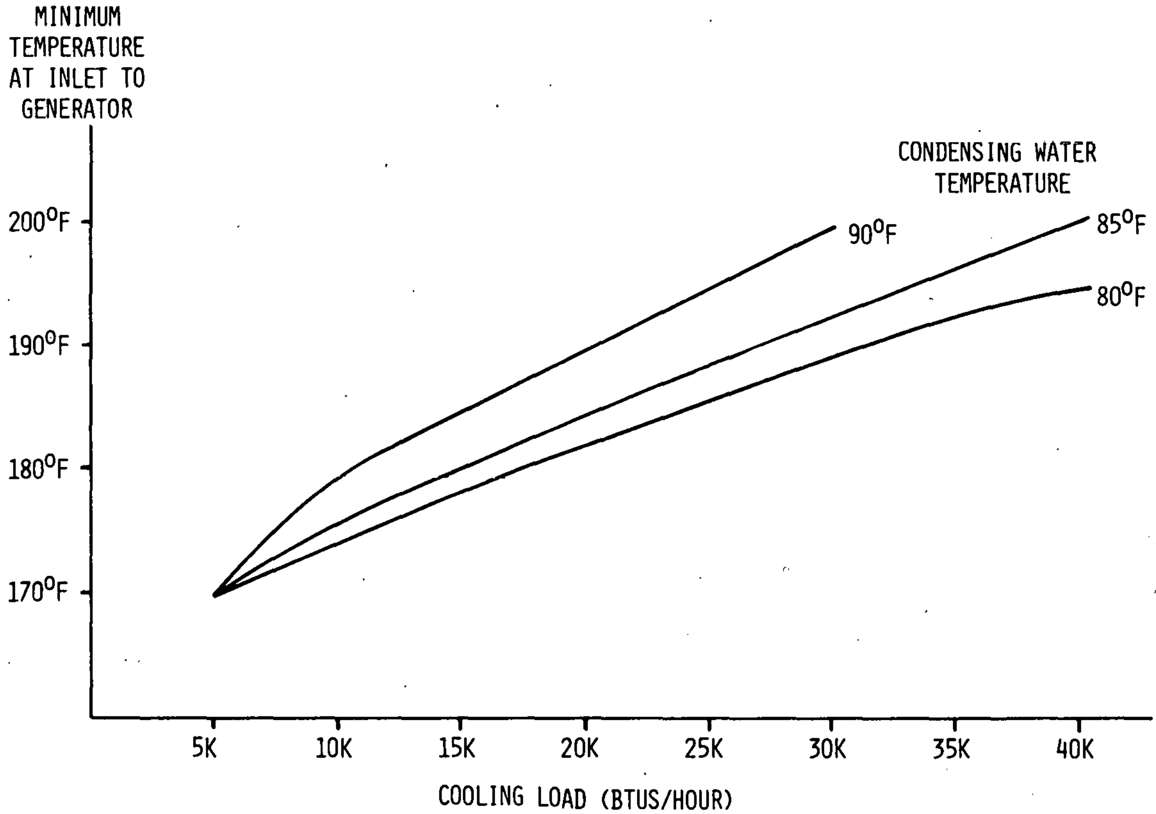

Figure 18. Inlet Temperature Required to Satisfy Cooling Load, ARKLA Model WF36 Three-Ton Absorption Unit, No Anticipation in Thermostat

With the presence of an auxiliary gas-fired boiler, the need for an auxiliary forced-air gas furnace is eliminated, reducing system cost accordingly (Table C-10).

By adding auxiliary heat to the storage loop to satisfy a heating load, some degradation in collector performance is to be expected. By the selection of an appropriate reset algorithm this degradation can be minimized.

Performance was simulated over one year in Atlanta under an algorithm that set the inlet temperature to the hot side of the solar coil as a function of heating load, solar coil effectiveness and mass flow on both sides of the coil: $T_{inlet} = 125^{o}F - 0.87\ T_{amb}$.

With this algorithm energy collected is reduced by only 1-1/2 percent. However the system pumps and blower consume additional power--some 325 kWHs per year.

Thus by eliminating the gas-fired forced-air furnace, there is an energy penalty that must be paid. In Atlanta this amounts to $11.88 per year at 1977 FEA rates. (Details in Tables C-3 and C-4.)

The initial cost of a residential forced-air furnace system is in the neighborhood of $500, yielding a payback period of 26 years in Atlanta. In the interest of simplicity, the recommendation is that the furnace be eliminated from the system. Parenthetically it should be pointed out that new developments in boiler design are in the direction of compactness, fast response and efficient heat transfer. European (and American) manufacturers are producing gas-fired boilers designed for residential applications (40,000 to 100,000 Btu/hour output) that nicely answer the design requirements of a combined solar heating and absorption cooling system for a residential application.

In an all-electric system the modest initial cost of the electric heating coil in the space heating loop does not preclude its use when a boiler is present in the system. Therefore, within the constraints of this contract work such a coil is recommended.

<u>Tradeoff V</u>--In the hydronic solar collector heating system designed for a single-family residence in Minnesota (System A), simulation

tradeoff studies indicate that differential temperature control of the storage tank vis-a-vis the heated space may introduce modest energy savings. The cost of implementing this control strategy is negligible, even with conventional control.

Therefore this storage strategy was also modeled and simulated within this hydronic heating/cooling system. Generally, with natural gas backup, the results are not as optimistic (Table 19). The parasitic power costs overpower the fuel savings in three regions.

Table 19. System C, Effect of Differential Temperature Control of Storage Tank Vis-a-vis Heated Space

REGION	REPRESENTATIVE CITY	SPACE HEAT AND DHW AUXILIARY ENERGY SAVINGS ($/YEAR)	PARASITIC POWER PENALTY ($/YEAR)	NET SAVINGS ($/YEAR)	PERCENT SAVINGS
MIDWEST	OMAHA	$ 2.81 ($11.70)	-$11.88	-$9.07 (+$ .18)	-4.6% --
NORTHEAST	NEW YORK	$ 4.52 ($12.86)	-$ 6.46	-$1.94 ($6.40)	-0.6% +0.7%
WEST	ALBUQUERQUE	$ 1.69 ($ 6.22)	-$ 2.26	-$ .57 ($3.96)	-0.5% +1.9%
SOUTH	ATLANTA	$ 2.98 ($ 8.23)	-$ 1.52	$1.46 ($6.71)	+1.0% +2.0%

AUXILIARY FUEL IS NATURAL GAS (ELECTRICITY IN PARENTHESES).
FEA 1977 RATES BY REGION ARE LISTED IN APPENDIX A.

As cited earlier, new developments in conventional control eliminate the need for an additional controller to implement differential temperature control of the storage tank. Thus the cost of the aquastat becomes a factor in considering absolute temperature control:

Absolute temperature control	\$42.44 (Cost of aquastat)
Differential temperature control	\$26.08 (Cost of extra sensor)

For the Midwest region (Omaha), absolute temperature control of the storage tank is recommended when natural gas is the auxiliary fuel. In all other regions differential temperature control is recommended.

With electric heat as backup, differential temperature control is recommended for all regions.

Tradeoff VI--The selected off peak storage management strategy evaluated is the simple outdoor reset strategy derived from the tradeoffs involving System A. It is probable that a function to predict cloud cover will enhance this simple algorithm. However the derivation of such a predictor is beyond the scope of this study.

The selected algorithm defines a minimum storage tank temperature for the heating season as a function of the system and building specifications. Auxiliary energy is added to storage during off peak hours to maintain this minimum temperature. Control is implemented by a simple outdoor reset thermostat.

Results of performance simulations indicate that the benefits of this strategy are in proportion to the total heating load (Table 20). Detailed results can be found in Table C-9.

In the southwest represented by Albuquerque the daytime auxiliary heating load is very modest--2.5 10^6 Btus in a year. Thus even though the off peak storage management strategy eliminates this auxiliary load entirely, the system savings are minimal (6 percent).

It appears that the level of savings generated by off peak storage charging in the climate represented by Albuquerque does not justify the cost of the auxiliary heating element.

Table 20. Benefits of Off Peak Storage Management Strategy in Four Regions

REPRESENTATIVE CITY	ANNUAL SPACE HEATING LOAD (WITH NIGHT SETBACK)	ANNUAL SAVINGS GENERATED BY OFF PEAK STRATEGY	ELECTRIC COST OFF PEAK
OMAHA	69.1×10^6	$ 406.21 OR 32%	3.43/PER KWH
NEW YORK	65.9×10^6	$1080.76 OR 44%	5.45/PER KWH
ALBUQUERQUE	46.7×10^6	$ 31.61 OR 6%	3.66/PER KWH
ATLANTA	49.4×10^6	$ 178.74 OR 24%	3.26/PER KWH

PEAK HOUR RATES ARE FOUR TIMES OFF PEAK RATES.

3. Recommended Control System

In the recommended control for this system, three applications are delineated:

- The heating/absorption cooling system for the south with natural gas backup: automatic thermostat night setback, economizer control, outdoor reset temperature control of storage, boiler in the solar loop.

- The heating/absorption cooling system for the midwest and west, with natural gas backup: automatic night setback, economizer control, absolute temperature control of storage, auxiliary furnace.

- The heating/absorption cooling system with electric energy backup: automatic night setback, economizer control, outdoor reset temperature control of storage, auxiliary heating elements in the solar loop, resistance auxiliary heating.

Components required to implement the recommended control are summarized below. (Detailed specificiations can be found in Appendix B.)

Component implementation requires:

- Six relays,

- An enthalpy controller,

- An advanced differential temperature controller,

- Three sensors,

- One aquastat,

- Two outdoor reset controllers, and

- An advanced two-stage thermostat with subbase and transformer.

Control mechanization components include:

- Five diverting valves,

- A damper motor,

- Three dampers with two linkages, and

- Four pumps.

When, as in the second option, the auxiliary boiler is omitted, the second stage heating function is taken over by an auxiliary gas furnace. Thus, an aquastat replaces one of the outdoor reset controllers, and one less valve is required.

In a system with electric energy as backup, there are indeed two outdoor reset controllers, one related to direct cooling and one to offpeak charging during the heating season. The capital investment relative to two auxiliary heating elements--one in the ARKLA cooling loop and the other in the air handler for space heating--is modest and precludes the need to combine these functions in a single boiler.

D. SYSTEM D, SOLAR ASSISTED HEAT PUMP SYSTEM FOR A SINGLE-FAMILY RESIDENCE

The work reported in this section was carried out under Solar Assisted Heat Pump Development, DOE Contract No. EG-77-C-03-1592, a joint contract between Lennox (prime contractor) and Honeywell. The DOE program manager, Mike Davis, and Bill Dollars, Lennox, principal investigator, are gratefully acknowledged for allowing the work to be reprinted here as it pertains so well to the objectives of this program.

This is a report on work in progress under the above contract. These are preliminary results, and are not intended to portray finalized conclusions.

1. Building and System Description

For these control simulations and evaluations, the baseline three-pump hydronic solar collector system has been selected. An advanced design dual-source four-ton heat pump for which performance characteristics are known, has also been selected.

By suppressing either the liquid or the air source of the heat pump performance profile, a liquid-to-air heat pump in series or an air-to-air heat pump in parallel can also be analyzed with reference to control options.

Figure 19 presents a schematic of this solar assisted heat pump system. The feature that distinguishes it from the residential systems previously analyzed is the great number of possible operating modes.

The optimization algorithm for the system using a two-speed dual-source heat pump must select from among seven heating possibilities:

1. Solar heat direct from the collector to the load.

2. Solar heat from storage to the load.

3. Air-to-air heat pump, low speed.

4. Water-to-air heat pump, low speed.

5. Air-to-air heat pump, high speed.

6. Water-to-air heat pump, high speed.

7. Electric resistance strip heaters.

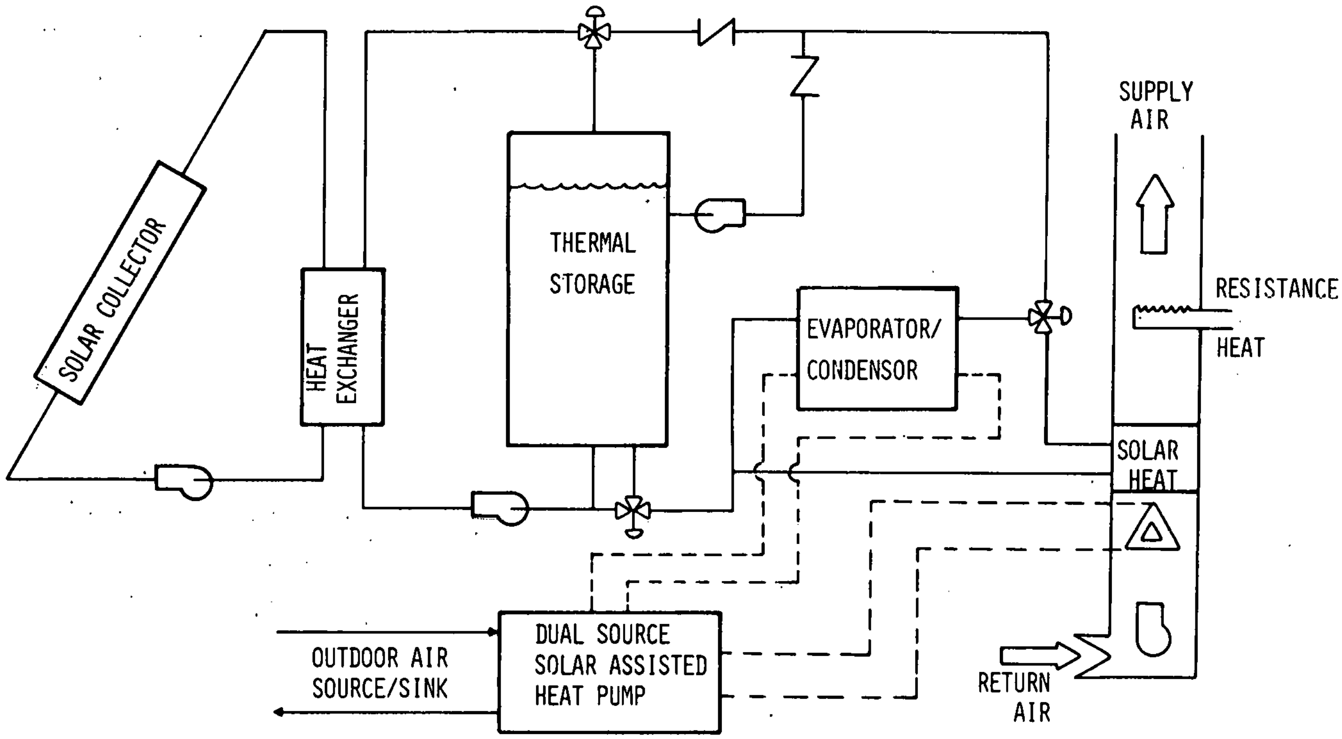

Figure 19. Solar Assisted Heat Pump System

Any of these modes could meet all of the load some hours of the year. Any two of the sources can be used in combination. Fortunately, (1) through (6) are mutually exclusive. That is, they cannot be operated at the same time.

Any of sources (1) through (6) could, under some circumstances, have less heating capacity than the instantaneous load. At the same time, some other source may have more than enough capacity. In such a case, the control system could turn on one of the over-

capacity sources for the remainder of the time. At any time it also may be possible for other combinations of sources to meet the load. The question is which combination uses the least electrical power including the parasitic power drains (fans and pumps) to meet the load.

To implement an optimal control philosophy we must know the machine duty cycle and net power consumption at any instant of time.

If two sources are to cycle on and off to meet the load, one must be less than the load and the other must be greater than the load. If both sources have less capacity than the load, they cannot meet the load in combination either. If both sources are greater in capacity than the load, then the optimal solution is to operate one or the other alone (whichever has the lowest required power) but not both.

The power, including parasitic pumps and fans, required to operate the devices is P_1 and P_2, respectively, so that:

$$P_{net} = DC_1 \; P_1 + DC_2 \; P_2$$

where DC_1 and DC_2 represent the duty cycle of each device and P_{net} is the net average power to service the load with the two devices working together.

For the case of one device cycling on and off alone, simply set P_2 to zero and the above algebra still holds.

To facilitate the control optimization procedure, the SUNSIM computer simulation has been programmed to select the combination

that generates the minimum power to meet the load each hour of the year. The simulation computes the available capacity for each device (whether it will be used or not), for each hour of the year, and uses the equation above to pick the best combination of devices (or single device alone) to use.

2. Tradeoff Evaluations

The purpose of these tradeoff evaluations is twofold:

- To implement an optimization algorithm based on minimum power consumption.
- To evaluate various heat pump configurations under the control of such an algorithm.

With this model and control algorithm, computer simulations have been completed for the dual-source heat pump, for an air-to-air heat pump parallel to the solar system, for a water-to-air heat pump in series with the solar system, for the solar system alone and also for electric resistance heat alone.

Six system configurations have been defined and modeled in the SUNSIM software. The dual-source heat pump model includes all seven heat sources. The other models are constructed by suppressing those sources not applicable:

System	Sources
Solar Only	1,2,7
Heat Pump Only	3,5,7
Parallel	1,2,3,5,7
Series	1,2,4,6,7
Dual-Source	1,2,3,4,5,6,7
Electric Resistance	7

The optimized control algorithm selects the best heat source among those available to the system. Analyses have been completed for three regions:

- The West represented by Phoenix,
- The Midwest region represented by St. Louis,
- The North Central region represented by Madison.

In each region the building load characteristics have been modified to match the four-ton design load of the heat pump described above. Within each region, collector area and storage volume have been varied to evaluate the effects on relative system performance.

Simulation results of performance over a typical year in both Madison and St. Louis indicate that dual-source heat pumps do generate the highest level of system performance vis-a-vis power requirements. As can be seen from Figures 20 and 21, relative seasonal COP is dependent on the collector area, but the superiority of the dual-source heat pump prevails.

Given typically sized collector areas, it is of interest to determine whether the additional cost of a dual-source heat pump is justified by the power reduction it introduces. Table 21 presents the payback computations for both Madison and St. Louis. Results indicate that payback is comparable when an optimal control algorithm is implemented.

For Phoenix, initial simulation results over a typical year recommended the parallel system with an air-to-air heat pump, regardless of collector area. However these simulations were based

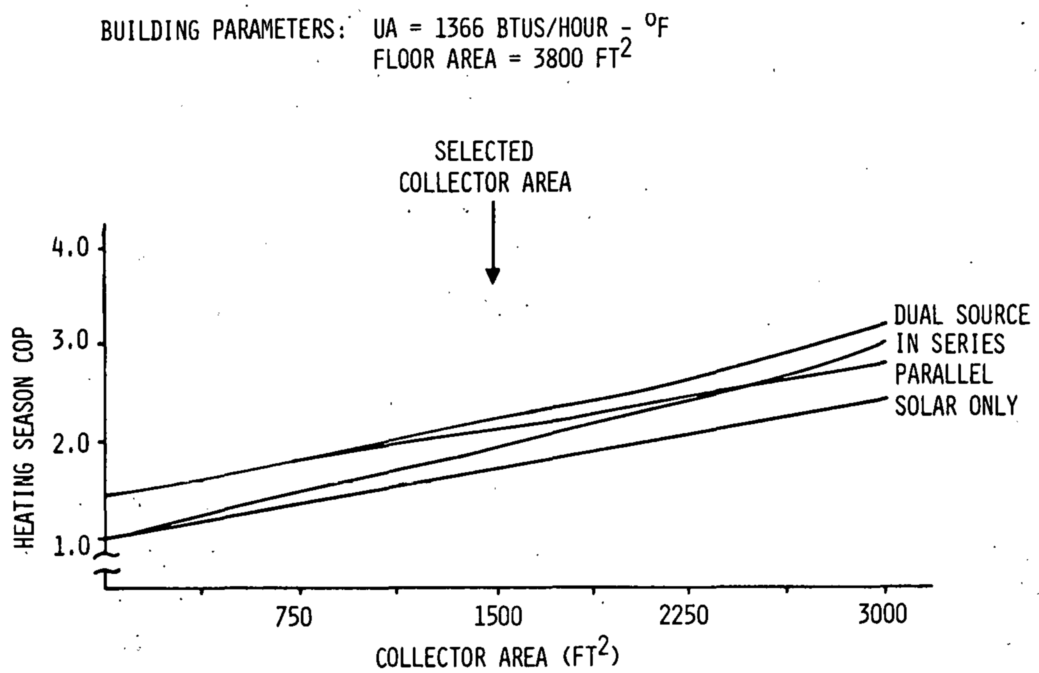

Figure 20. Heat Pump System Coefficient of Performance, Madison, Wisconsin

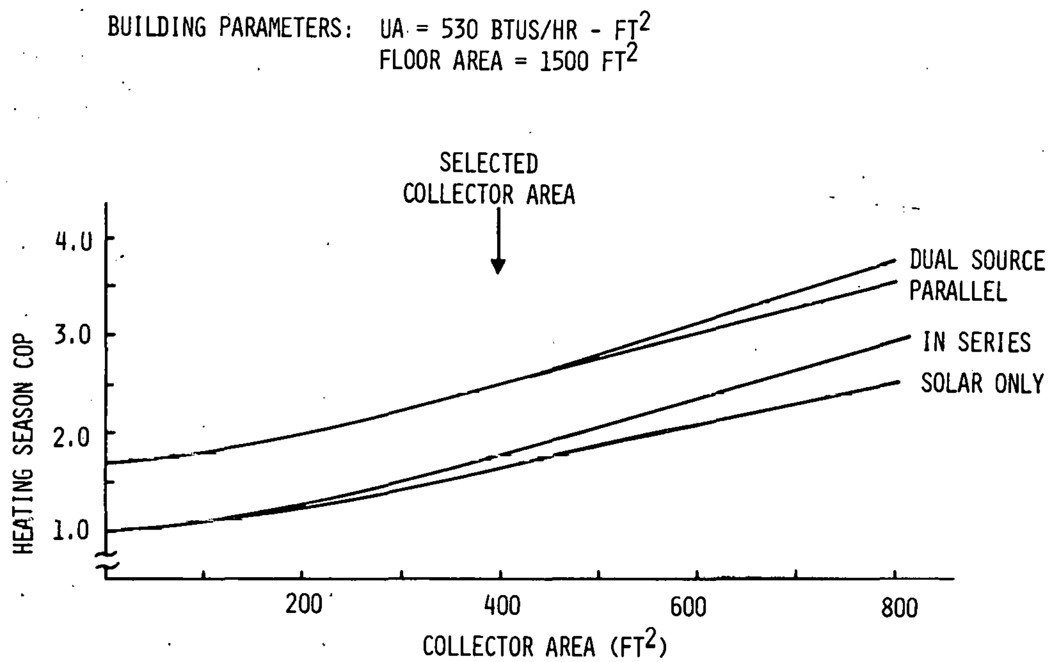

Figure 21. Heat Pump System Coefficient of Performance, St. Louis, Missouri

on a 35^oF temperature set in the storage tank, which is appropriate for the Madison and St. Louis climates.

Table 21. Simulation Results and Analyses, Heat Pump Options in Two Regions

	INITIAL COST DIFFERENCE OVER CONVENTIONAL SOLAR SYSTEM	ADDITIONAL FIRST YEAR ENERGY SAVINGS OVER CONVENTIONAL SOLAR SYSTEM	FIRST YEAR ELECTRICITY SAVINGS OVER CONVENTIONAL SOLAR	PAYBACK PERIOD FOR HEAT PUMP SUBSYSTEM
MADISON, WISCONSIN	.4 FT^2 COLLECTOR AREA/FT^2 FLOOR SPACE			
AIR-TO-AIR HEAT PUMP IN PARALLEL WITH SOLAR	$3237	8,600 KWHS	$318.20	11 YEARS
WATER-TO-AIR HEAT PUMP IN SERIES WITH SOLAR	$3539	5,000 KWHS	$185.00	22 YEARS
DUAL SOURCE HEAT PUMP	$3998	10,500 KWHS	$388.50	11 YEARS
1977 FEA ELECTRICITY RATE $.037/KWH PROJECTED ANNUAL ESCALATION RATE + .8% ANNUAL DISCOUNT RATE ABOVE INFLATION 2% (APPENDIX A)				
ST. LOUIS, MISSOURI	.3 FT^2 COLLECTOR AREA/FT^2 FLOOR SPACE			
AIR-TO-AIR HEAT PUMP IN PARALLEL WITH SOLAR	$3237	5,000 KWHS	$180.00	22 YEARS
WATER-TO-AIR HEAT PUMP IN SERIES WITH SOLAR	$3539	1,100 KWHS	$ 39.60	NONE
DUAL SOURCE HEAT PUMP	$3998	4,900 KWHS	$176.40	29 YEARS
1977 FEA ELECTRICITY RATE $.036/KWH PROJECTED ANNUAL ESCALATION RATE + .3% ANNUAL DISCOUNT RATE ABOVE INFLATION 2% (APPENDIX A)				

Figure 22 is a plot of the storage tank and ambient temperatures in Phoenix during five days in January. The dominant effect of the control algorithm (which includes a storage setpoint of 35^oF) is to select the water source at night since it generates a higher COP. The parallel system, however, switches to the air pump as soon as the storage temperature drops below 70^oF. Further simu-

lation runs indicate that performance of the dual-source heat pump is superior when a 70°F setpoint is maintained in the storage tank.

What appears to surface clearly from these control evaluations is the complexity of the control logic required for effective operation of a solar assisted heat pump system.

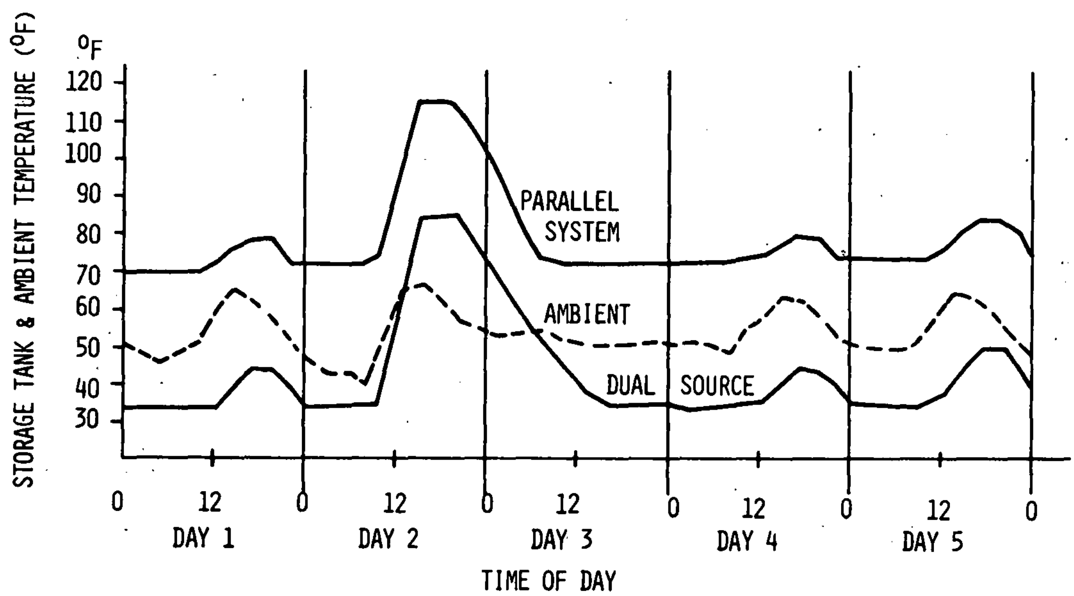

Figure 22. Simulation Results, January Profile in Phoenix, Storage Tank Setpoint of 35°F

3. Recommended Control System

It appears that effective control of this solar assisted heat pump system requires an algorithm based on ambient temperature and the storage tank temperature.

A "map" is being developed at Honeywell to select the appropriate control mode as a simple function of ambient and storage temperatures. A control system consisting of reset controllers, differential temperature controllers, relays, aquastats and other state-of-the-art components is expected to implement control satisfactorily.

The power minimization control algorithm defined here for System D could also be implemented by some level of microprocessor control (see Part I, Survey of Advanced Controllers).

E. SYSTEM E, COMMERCIAL BUILDING HYDRONIC SOLAR HEATING/COOLING SYSTEM

1. Building and System Description

The commercial building defined for this study is a typical new office building. Studies have shown that schools and offices account for seven percent of the energy use nationwide, and for purposes of this study, the 12-month operating pattern of a typical office building renders it most suitable.

Drawing on results of recent work completed at the Honeywell Energy Resources Center and cited previously, the typical office building has been defined: a three-story building with 30,000 square feet of floor space. Its construction varies somewhat from region to region (Table 22).

The building envelope heat transfer parameters become:

3221 Btus/hr-^{o}F for the North Central region,

2878 Btus/hr-^{o}F for the Northeast,

2878 Btus/hr-°F for the West,

3221 Btus/hr-°F for the South.

Heat transfer due to air infiltration is assumed to be 15 percent of the envelope heat transfer in each case. The solar load on the windows is also modeled using data from Table 22.

Based on the previous study, an interior heat gain for these office buildings has also been defined:

- One person per 50 square feet of office space, each person generating 250 Btus of sensible heat and 200 Btus of latent heat per hour and consuming 0.5 gallons of hot water per day.

- High efficiency lights requiring two watts power per square foot of office space.

- A five-day week building operating schedule from 6 a.m. to 10 p.m. Figures 23 through 26 present the hourly schedules.

For all regions the minimum ventilation heating and cooling load is 7 cfm per person, or 1400 cfm for the 7500-square-foot office building.

The solar heating/cooling system model for this office building reflects previous design work by the Energy Resources Center.

The solar subsystem portion closely resembles the residential heating/cooling system configuration (Figure 27). The size of the office building has been scaled (7500-square-feet to fit the 25-ton ARKLA absorption unit.)

Table 22. Office Building Construction Characteristics

CHARACTERISTIC	NORTHEAST	NORTH CENTRAL	SOUTH	WEST
SIZE - FT^2/FL	10,000	10,000	10,000	10,000
DIMENSIONS	100 FT. X 100 FT.	100 FT. X 100 FT.	100 FT. X 100 Ft.	100 FT. X 100 FT.
NO. OF FLOORS	3	3	3	3
HEIGHT OF STORY	12 FT.	12 FT.	12 FT.	12 FT.
FOUNDATION TYPE	4 IN. SLAB	4 IN. SLAB	4 IN. SLAB	4 IN. SLAB
ROOF TYPE (FLAT)	BUILD UP, METAL DECK	BUILD UP, METAL DECK	BUILD UP, METAL DECK	BUILD UP, METAL DECK
INSULATION	2 IN. RIGID	2 IN. RIGID	2 IN. RIGID	2 IN. RIGID
EXT. GLASS (FT^2)	4320	4320	4320	4320
TYPE	1 IN. INSULATED	1 IN. INSULATED	1 IN. INSULATED	1 IN. INSULATED
EXT. DOORS (FT^2)	160	160	160	160
TYPE	1 IN. INS. GLASS	1 IN. INS. GLASS	1 IN. INS. GLASS	1 IN. INS. GLASS
EXT. WALL:	METAL CURTAINWALL	MASONRY	METAL CURTAINWALL	MASONRY
INSULATION	2 IN. RIGID	1 IN. RIGID	2 IN. RIGID	1 IN. RIGID
INT. FINISH	GYPSUMBOARD	NONE GYPSUMBOARD	GYPSUMBOARD	GYPSUMBOARD
AREA (FT^2)	10,080	10,080	10,080	10,080

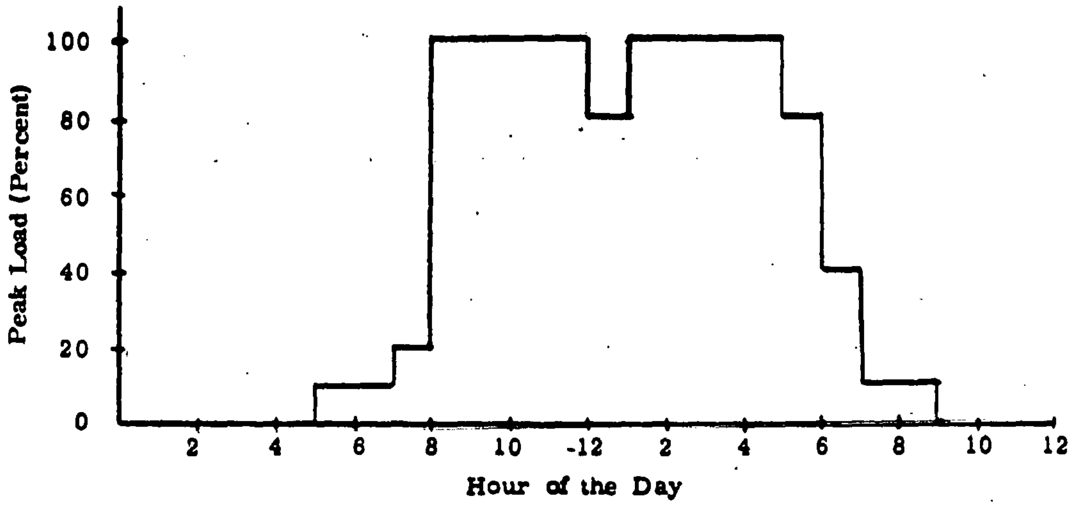

Figure 23. Office Building People Load Schedule

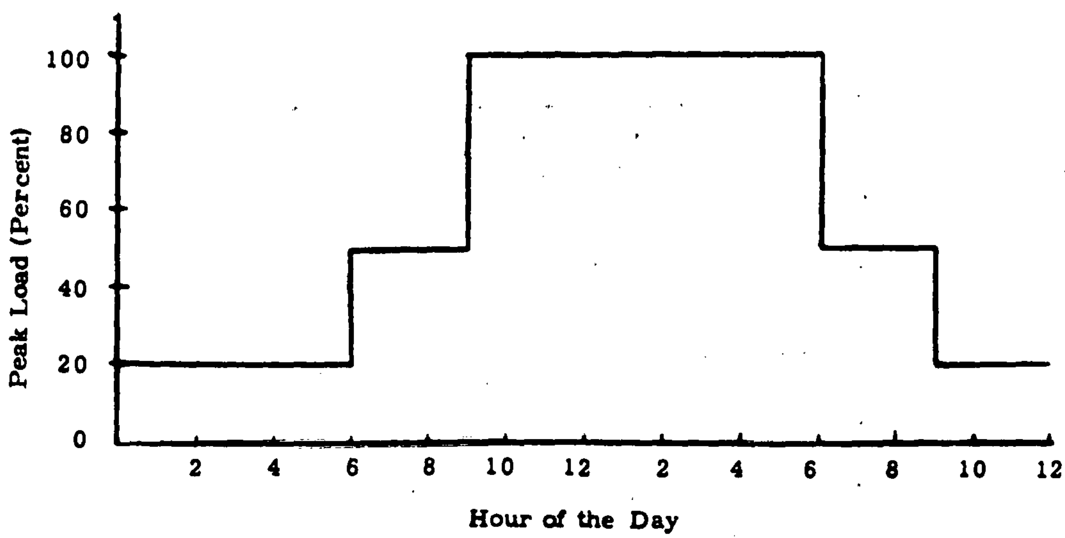

Figure 24. Office Building Light Schedule

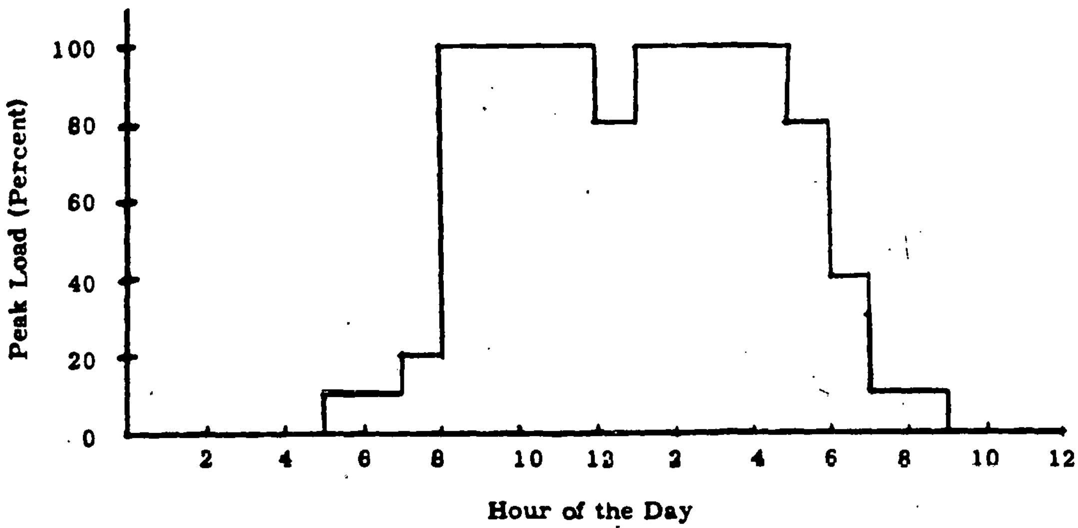

Figure 25. Office Building Hot Water Schedule

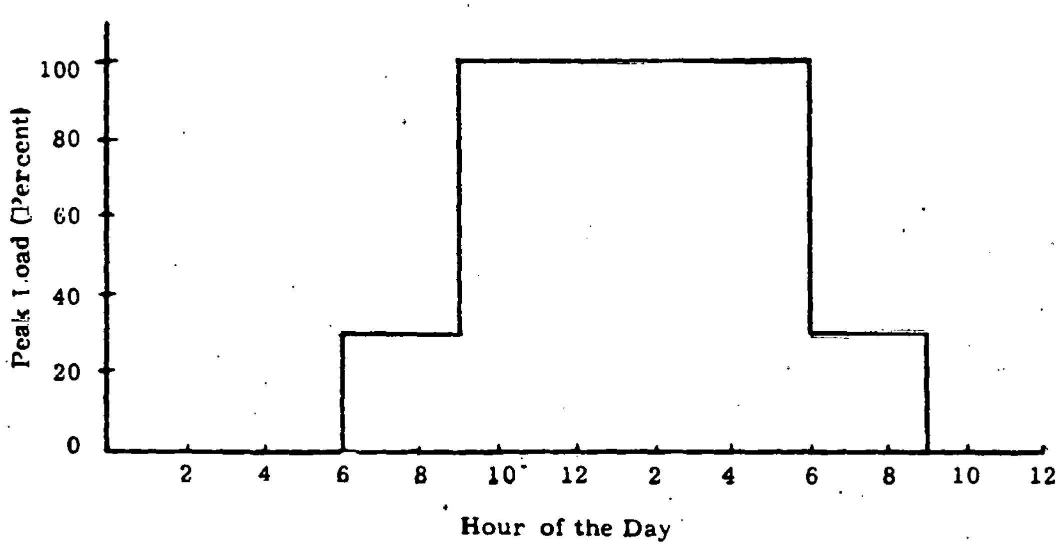

Figure 26. Office Building Ventilation Schedule

The HVAC subsystem is the variable air volume (VAV) system recommended in the Mixed Strategies study cited above, involving several zones on each floor. The assumption is made that simultaneous heating and cooling loads in the building can be adjusted through the appropriate control of dampers and valves within the HVAC portion of the system:

- Heat is supplied by conventional sources through baseboard perimeter radiators, controlled by zone thermostats, throughout the year.
- During the heating season only heat from solar sources is used to preheat ventilation and infiltration air while the VAV system is on. Auxiliary energy is provided when no heat is available from the solar system.
- In both the heating and cooling seasons, the economizer may be used for cooling.
- Otherwise cooling is supplied by the absorption cooling unit through the several coils that interface with the VAV system.
- The absorption cooling unit can handle the building design cooling load, with auxiliary energy provided to the unit as necessary.

The components of the solar system are sized as follows:

- 3000 square feet of Lennox flat plate collectors (166 two-cover collectors).

- Collectors mounted two in series with 0.5 gpm 25 percent ethylene glycol solution flowing through each collector and 42 gpm through the heat exchanger.

- A 4500-gallon storage tank with a surface heat loss of 20 Btus/hr-^{o}F to unconditioned space.

- A storage loop flow of 90 gpm for both the heating and cooling season.

- A heat exchanger with an effectiveness of 0.6 between the collector and storage loops.

- An ARKLA model WFB300 25-ton absorption cooling unit (Figure 28):

  - Design hot water flow, 90 gpm (proportionally controlled),
  - Design chilled water flow, 9.8 gpm,
  - Design hot water inlet temperature, 195^{o}F,
  - Design chilled water outlet temperature, 45^{o}F,
  - Design delivered capacity, 306,000 Btus/hour.

- Six ARKLA vertical fan coil filter assemblies model VFCF 48-109:

  - Total cooling capacity, 288,000 Btus/hour,
  - Total heating capacity, 654,000 Btus/hour,
  - Rated water flow, 9.6 gpm/unit,
  - Rated air flow, 1600 cfm/unit,
  - Effectiveness at design conditions, 0.7.

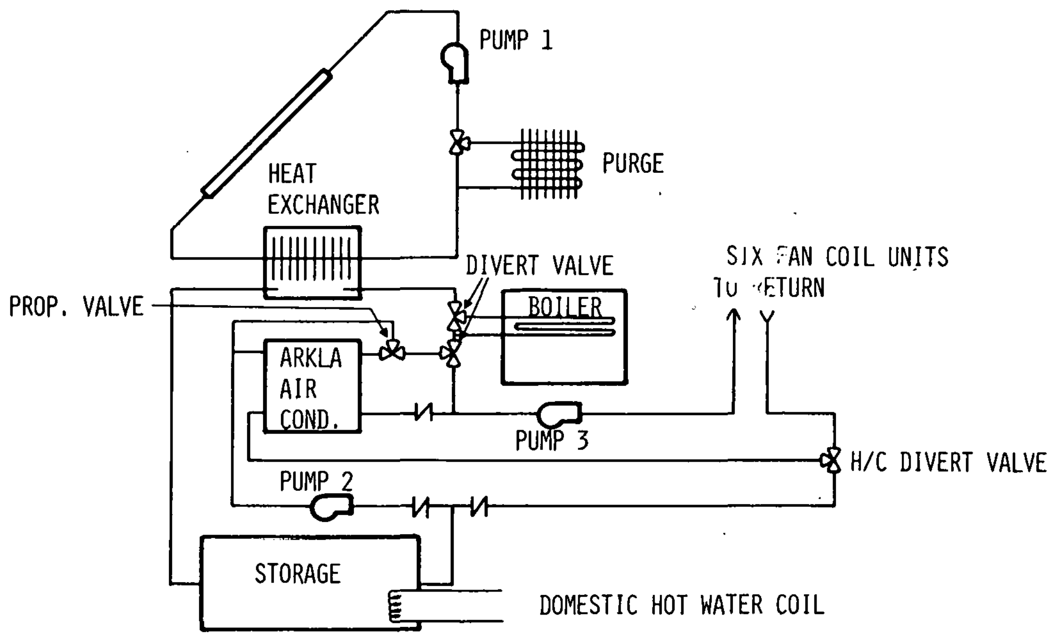

Figure 27. Hydronic Solar Collector Heating/Cooling System for a Commercial Building

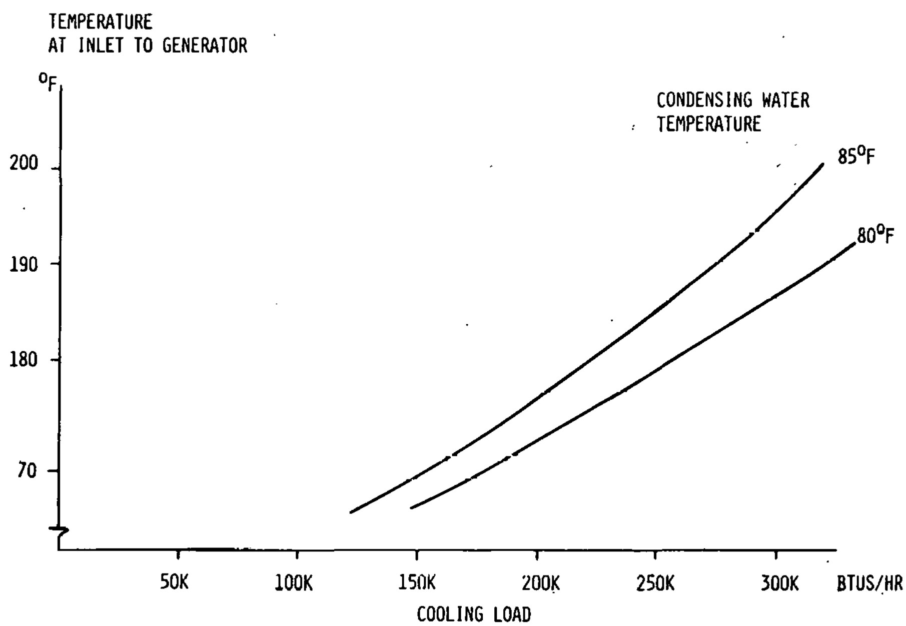

Figure 28. Performance of the 25-Ton ARKLA Absorption Air Conditioner

2. Tradeoff Evaluations

Two sets of tradeoff evaluations were defined, one set applying to a system with auxiliary gas as the backup heating energy source and one set relating to an all-electric system. All tradeoff studies were completed for four regions.

Tradeoff I--In a gas-fired auxiliary subsystem, the question to address involves system configuration. Is it more economical to employ a gas-fired boiler as a booster in the solar/absorption loop, or to eschew the boiler in favor of a gas-fired forced-air furnace and conventional compressor air conditioning? The answer appears to be region-dependent. Simulations and analyses relating to this tradeoff rested on certain assumptions, namely:

- A gas fired boiler system with a seasonal performance of 60 percent in all regions.
- The gas-furnace system with a seasonal efficiency of 60 percent in all regions.
- The conventional air conditioner with a COP of 4.0 in all regions.

Additional pumping power requirements were estimated as follows:

Collector Loop Pump 1	1 hp	750 watts
Storage Loop Pump 2	2 hp	1700 watts
Delivery Loop Pump 3	3/4 hp	600 watts
Air Handler Blowers (6)	1/2 hp	800 watts

Commercial energy rates--both gas and electric--are delineated by region in Appendix A. Table 23 presents the annual energy costs

Table 23. Hydronic Solar Heating/Cooling System for a Commercial Building, Gas Auxiliary Heating/Cooling Configuration and Control

TRADEOFF I	STORAGE LOOP GAS-FIRED BOILER	AUXILIARY GAS FURNACE	AUXILIARY CONVENTIONAL AIR CONDITIONER	TOTAL HEATING & COOLING COSTS $/YEAR		HEATING/COOLING LOADS NOT SATISFIED BTUS X 10^6/YEAR
OMAHA						
CONVENTIONAL		X	X	$ 846	-	-/-
FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	X			$ 854	+ 1%	.8/34.6
RESET ALGORITHM	X			$ 982	+16%	-/-
NEW YORK						
CONVENTIONAL		X	X	$1394	-	-/-
FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	X			$1560	+12%	-/20.4
RESET ALGORITHM	X			$1723	+24%	-/-
ALBUQUERQUE						
CONVENTIONAL		X	X	$ 527	-	-/-
FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	X			$ 508	- 4%	.2/13.4
RESET ALGORITHM	X			$ 591	+12%	-/ .1
ATLANTA						
CONVENTIONAL		X	X	$ 879	-	-/-
FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	X			$ 960	+ 9%	-/64.7
RESET ALGORITHM	X			$1236	+41%	-/ .5

DETAILED COMPUTATIONS IN APPENDIX C

under each of the two auxiliary options. With the boiler, performance was simulated with fixed temperature setpoints in the storage loop and with outdoor temperature reset algorithms. Detailed simulation results are shown in Tables C-11 throuth C-14. From these it is clear that a fixed setpoint of 165^{o}F in the cooling season is not adequate to handle the cooling load in any region.

The initial cost of an auxiliary air conditioner plus an auxiliary gas furnace as compared to the cost of a boiler in the storage loop recommends the latter if the cooling load can be reasonably satisfied.

It appears that the boiler option is suitable for at least two regions--the midwest and the southwest--if the storage loop setpoints are effectively controlled, namely:

- Fixed setpoint, implemented by an aquastat, for the heating season.
- Linear outdoor reset algorithm for the cooling season.

Depending on first costs of the gas furnace and conventional air conditioner vis-a-vis the cost of a boiler, it may be that the former are the appropriate selections for the Northeast and the South.

Tradeoff II--In an all-electric system, the auxiliary heating elements introduce more modest first costs than those associated with a gas system. Auxiliary space heat is provided by electric resistance heating elements, as is storage charging during off peak periods. Therefore component configuration tradeoffs are omitted.

Included are the various options associated with off peak storage management strategies. As in the residential system evaluations, an off peak scenario was deinfed: Off peak rates in effect 6 p.m. to 6 a.m. and peak rates multiplying by a factor of 4.

The tradeoff analysis results are detailed in Table C-15. A heating season linear storage temperature limit algorithm based on outdoor temperature was defined for each region. Not unexpectedly, by applying this algorithm energy costs were reduced, though collector performance was somewhat degraded. These results are presented in Table 24.

Table 24. Hydronic Solar Heating/Cooling System for a Commercial Building, All-Electric System Off Peak Heating Season Strategy

TRADEOFF II	COLLECTOR PERFORMANCE BTUS 10^6/YEAR	%	ANNUAL HEATING & COOLING ENERGY COSTS $/YEAR	%
OMAHA				
NO STRATEGY	346.2	-	$ 5,582	-
OFF-PEAK STORAGE CHARGING	318.6	-8%	$ 4,260	-24%
NEW YORK				
NO STRATEGY	314.9	-	$10,351	-
OFF-PEAK STROAGE CHARGING	296.1	-6%	$ 8,379	-19%
ALBUQUERQUE				
NO STRATEGY	556.0	-	$ 2,355	-
OFF-PEAK STORAGE CHARGING	537.6	-3%	$ 2,195	- 7%
ATLANTA				
NO STRATEGY	430.5	-	$ 4,109	-
OFF-PEAK STORAGE CHARGING	402.6	-6%	$ 3,395	-17%

DETAILED COMPUTATIONS IN APPENDIX C

3. Recommended System

As in the case of the residential heating/cooling system, recommendations for this commercial system application are regionally based.

For the southwest and midwest regions--represented by Albuquerque and Omaha, respectively--a gas-fired boiler in the storage loop is the recommended configuration. For Omaha the energy costs are virtually the same with a gas-fired boiler or with a conventional auxiliary system. However, the initial cost benefit in selecting the storage loop boiler supports this recommendation. Aquastat control of the loop is recommended for the heating season, and outdoor reset control for the cooling season. From a thermostat point of view, this is equivalent to a single heating stage and two-stage cooling, with automatic changeover.

In the other two regions the recommended gas backup system does not include the gas fired boiler. Rather, an auxiliary gas furnace and conventional cooling are recommended. Relative to the thermostat, there are four stages of climate control:

- First stage heating - solar/storage,
- Second stage heating - gas furnace,
- First stage cooling - solar/absorption, or economizer,
- Second stage cooling - conventional compressor.

In an all-electric system auxiliary resistance heat and conventional compressor cooling are recommended for all four regions. For the thermostat, this means two stages of heating and two stages of cooling. There is no boiler in the storage loop.

Off peak strategies, implemented by an outdoor reset controller, are recommended for all regions. The recommended storage charging mechanism is a heating coil in the storage tank, eliminating any pumping power requirements.

F. SYSTEM F, COMMERCIAL BUILDING SOLAR CONCENTRATING COLLECTOR SYSTEM FOR HEATING/COOLING APPLICATION

1. Building and System Description

The commercial building defined here is the new Honeywell General Offices building, located in Minneapolis and now in its final stages of construction (Figure 29). This is an eight-story building with 100,000 square feet of floor space and a predicted weekday occupancy of 500 people.

Energy conservation concerns dictated the building design as well as building operation strategies:

- Thermal transmittance values on walls and roof are 10 percent more stringent than 1976 Minnesota Building Code Requirements and 30 percent more stringent than the ASHRAE 90-75 Standard.

- Special sunshades over the south and west exposed windows are dimensioned and positioned to admit direct sunlight in winter and restrict it in summer.

- The lighting system throughout minimizes cooling load and electric demand: all fluorescent lighting with high power factor ballast, lighting load not to exceed 1.5 watts per square foot.

- True variable air volume air handler.

Solar Collector System--The solar collector system features a 20,250-square-foot field of trough type concentrating collectors designed and built at the Honeywell Energy Resources Center. These are located on a steel support structure built above a five-story parking ramp (Figure 29). The collected thermal energy is piped underground over 300 feet from the parking ramp to the solar HVAC equipment located in a first floor mechanical room in the new office building.

Figure 29. Honeywell General Offices Solar Collector Field on Parking Ramp

The solar collector system has been sized to meet the peak cooling load of the new building (200 tons) and thus it will demonstrate the capability of solar energy to offset the peak electrical power demands in high rise commercial buildings, demands which typically occur under summer midday air conditioning loads.

The collector field consists of 42 rows. Each row individually tracks the sun's elevation with a closed loop sun tracking system comprised of a photosensitive sensor, motor drive electronics and an ac motor/brake assembly. The electronics also contain a temperature sensing circuit, which offpoints the collector if the absorber outlet pipe exceeds 400°F.

The solar system (Figure 30) employs two major flow loops. The primary or collector loop supplies solar energy at 390 gpm using heat transfer oil Therminol 44 as the energy transport fluid.

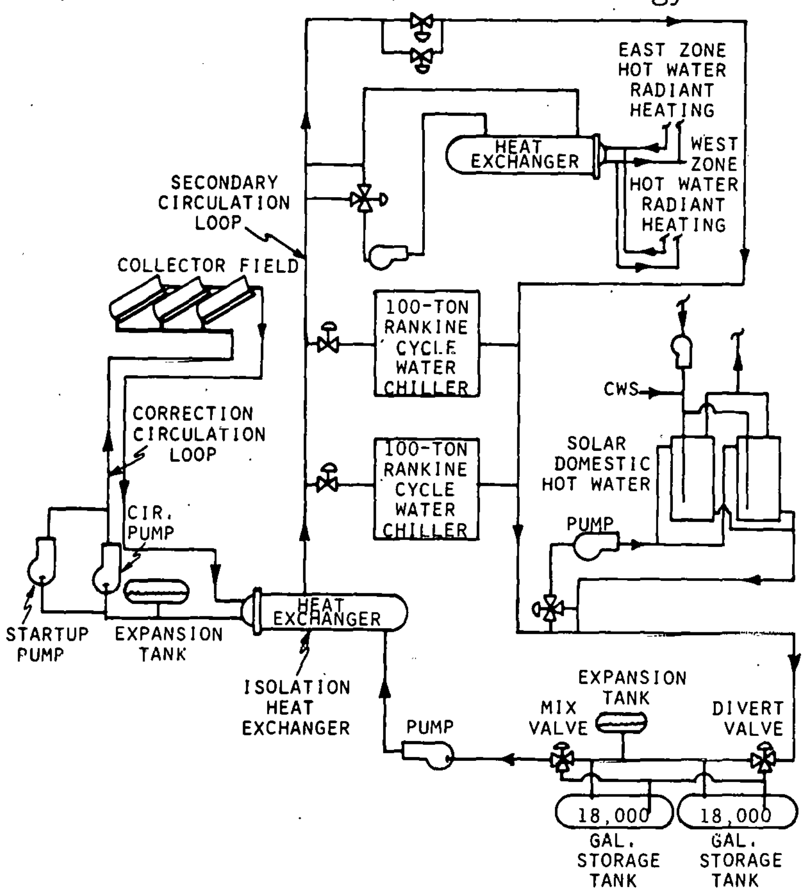

Figure 30. Solar System Flow Schematic

The secondary flow loop employs Caloria HT 43 (460 gpm) as the heat transport agent carrying solar energy to the building solar heating, cooling and hot water systems and the high temperature thermal storage subsystem. In the mechanical room, thermal energy is converted to useful HVAC energy in response to the building load demand. Any excess solar energy is piped to two underground thermal storage tanks where it is stored for later use. Expansion of the heat transfer fluids is taken up by a 750-gallon tank in the collector loop and a 4,000-gallon tank in the secondary loop. The isolation heat exchanger is provided to allow different fluids, pressures and flow rates to be used in the two flow loops.

The entire solar HVAC system is automatically controlled by conventional industrial and commercial electropneumatic controllers, relays, valves and sensors. A Honeywell computer-based building management system continuously monitors the solar system operation and provides status and alarm information to building operations personnel and to a 24-hour security guard station.

Solar Cooling System--The solar cooling system features two Rankine cycle power systems which drive two 100-ton water chillers. The water chillers are conventional open centrifugal units that provide chilled water for a distributed air cooling system. The Rankine cycle turbine is mechanically coupled to the chiller compressor motor by means of a gearbox containing a mechanical over-running clutch. When solar energy is not available, the electric motor provides a 100 percent backup to the solar driven Rankine turbine at a COP of 5.46.

Each Rankine cycle engine has a design point capacity of 85 hp. This will be available at a solar fluid flow rate of 230 gpm at 300°F and a condensing water flow rate of 330 gpm at 85°F. The

solar Rankine engine is merely a low temperature steam generator cycle whose working fluid is refrigerant R113 instead of water. Solar energy is transferred to the refrigerant in a preheater and evaporator, where the R113 is evaporated. The R113 vapor (275^{o}F, 133 psi) is expanded through the turbine to 178^{o}F and 10.3 psi, producing the required net output shaft power of 85 hp. Following refrigerant condensation in a condensing heat exchanger, the R113 is pumped back to the preheater and evaporator where the Rankine cycle is repeated. The Rankine cycle power system for this program has been designed, fabricated and assembled at Barber-Nichols Engineering in Denver, Colorado.

Solar Heating System--The solar heating system consists of a separate circulating loop that pumps heated oil at 182 gpm into a shell and tube heat exchanger (Figure 30). Solar energy heats the return hot water in the building radiant baseboard heating system. The building east and west zone radiant heating water temperatures are controlled and reset by an outdoor air thermostat. After leaving the solar heat exchanger, the water flows through a conventional steam converter for each zone, which will provide any auxiliary energy required to meet the hot water supply temperature requirement. The solar controls are designed such that any solar energy is provided in priority to the steam energy.

Solar Domestic Hot Water System--Domestic hot water required for the building is heated entirely by solar energy. This is accomplished through a separate circulating loop and two 120-gallon "double-wall" hot water heaters (Figure 30). The heated water is circulated to all eight floors during occupied hours only.

Thermal Storage System--The storage system consists of two 18,000-gallon underground steel tanks containing a mixture of Caloria HT

43 and small rocks. Incorporation of the rocks in the storage tank added to the tank thermal capacitance and reduced the costs of the oil for the storage system. The thermal capacitance of the two 18,000-gallon tanks is 150,000 Btu/^{o}F. The well insulated storage tanks are located five feet underground and 70 feet away from the new building under the ground level parking lot shown in Figure 29. The control system provides a capability to charge and discharge and sequence one or both of the thermal storage tanks so as to maximize both solar collection efficiency and solar conversion efficiency in the solar heating and cooling subsystems.

2. Control Tradeoff Evaluations

The baseline control system design incorporates accepted energy conservation strategies for commercial buildings:

- Weekend shutdown of ventilation and cooling systems.
- Night shutdown for 12 hours during the heating season.

The control tradeoff evaluations all relate to the advanced Rankine cycle cooling system, solar assisted, for which recognized control standards have not heretofore been established.

A detailed model of the new building heating, cooling and hot water loads was entered into the SUNSIM software.

The model includes exterior thermal transmission, air infiltration and ventilation, internal people and equipment loads, outdoor air economization and hourly solar loading on the building windows. The simulation model of the solar HVAC system includes collector field performance, thermal dynamics of the storage system, cooling

tower performance as a function of load and ambient conditions and detailed operation characteristics of the solar heating, cooling and hot water subsystems under all possible modes of operation.

A statistical analysis of Minneapolis weather tapes from 1949 to 1958 indicates that the summer and winter weather, as well as the available solar insolation, for the year 1956 is the most representative of the ten-year average of these parameters, and thus the weather tape for 1956 was used for solar system performance predictions (see Table 25).

Table 25. Honeywell General Offices System Annual Heating/Cooling Performance, 1956 Minneapolis Weather Tape

PARAMETER	VALUE
SOLAR ENERGY AVAILABLE	6.85 BILLION BTU
SOLAR ENERGY COLLECTED	2.68 BILLION BTU
BUILDING HEATING LOAD	2.32 BILLION BTU
SOLAR SUPPLIED HEATING	1.22 BILLION BTU
SOLAR HEATING CONTRIBUTION	53 PERCENT
BUILDING COOLING LOAD	0.72 BILLION BTU
SOLAR SUPPLIED COOLING	0.61 BILLION BTU
SOLAR COOLING CONTRIBUTION	84 PERCENT
COOLING ELECTRIC POWER	3884 KW-HR
BUILDING HOT WATER LOAD	73 MILLION BTU
SOLAR SUPPLIED HOT WATER	73 MILLION BTU
SOLAR HOT WATER CONTRIBUTION	100 PERCENT
HEATING FROM COLLECTORS	313 HOURS
HEATING FROM STORAGE	976 HOURS
COOLING FROM COLLECTORS	356 HOURS
COOLING FROM STORAGE	327 HOURS
CHARGING STORAGE	1598 HOURS
AUX. HEATING	44 HOURS
AUX. COOLING	62 HOURS
DORMANT HOURS	4760 HOURS
FULL CHARGED IN STOW	386 HOURS

With these software models three sets of control tradeoff evaluations were completed.

Tradeoff I--In the survey of strategies the substantial energy savings possible in commercial buildings through enthalpy, or economizer, control were cited.

The initial control tradeoff simulations for this System F evaluated proportional enthalpy control with and without solar. Table 26 presents the results for this large building in Minneapolis. With conventional water chiller cooling, this economizer control reduces the annual electricity draw by 20,320 kWHs, or $736 at 1977 commercial rates (Appendix A). With the Rankine cycle solar assisted cooling system, there is an additional annual savings electricity draw of 24,728 kWHs, or $896. There is a net decrease in seasonal collector performance due to enthalpy control. However this degradation is overshadowed by the electric power savings.

These savings appear to justify the cost of sensors and damper control required to vary the ventilation air intake in this variable air volume HVAC system. Economizer control is therefore selected as a cost-effective strategy within this system.

Tradeoff II--The effect of the water chiller temperature on solar system performance was examined in a second set of tradeoff evaluations.

Traditionally building cooling systems employ water chillers that are controlled to provide a chilled water supply at a constant temperature (46°F is typical). By resetting this supply temperature upwards as a function of load, the thermodynamic efficiency of the water chiller equipment is improved significantly yielding a net decrease of 718 kWHs in power requirements over one year, according to the simulation predictions (Table 27). These electricity savings amount to $29. The cost of implementing this strategy is trivial within the large complex HVAC control system.

Table 26. Influence of Economizer and Solar HVAC on Cooling, Honeywell General Offices System

ANNUAL PERFORMANCE PARAMETER	NO ECONOMIZER NO SOLAR	NO ECONOMIZER WITH SOLAR	WITH ECONOMIZER NO SOLAR	WITH ECONOMIZER WITH SOLAR
COOLING LOAD, BTU	1.265×10^9	1.265×10^9	0.7222×10^9	0.7222×10^9
SOLAR COOLING, BTU	--	1.045×10^9 (82.6%)	--	0.5896×10^9 (81.6%)
COOLING DRAW, KW-HR	49,650	10,320	29,330	4,602
COLLECTED ENERGY, BTU	--	3.032×10^9	--	2.712×10^9
COOLING FROM COLLECTORS, HOURS	--	857	--	366
COOLING FROM STORAGE, HOURS	--	621	--	314
STORAGE CHARGING, HOURS	--	1165	--	1609
STOW, HOURS	--	199	--	375

Table 27. Effects of Chilled Water Reset

ANNUAL PERFORMANCE	NO SOLAR W/O RESET	NO SOLAR W. RESET	SOLAR W/O RESET	SOLAR W. RESET
COLLECTED ENERGY, 10^9 BTU	0	0	2.712	2.6878
COOLING LOAD, 10^9 BTU	.7222	.7222	.7222	.7222
SOLAR COOLING, 10^9 BTU	0	0	.5896	.6087
PERCENT COOLING FROM SOLAR	0	0	81.64	84.28
COOLING ELECT. POWER, KW-HR	29,333	25,900	4602	3884

Tradeoff III--A solar assisted Rankine cycle cooling unit has distinctive characteristics that essentially demand variable speed control of the system under some conditions.

In Figure 31 these characteristics are illustrated. Variable speed operation of a Rankine engine is significantly more efficient (for the thermal and electrical energy) than constant speed operation. During constant speed operation the water chiller turning vanes control chiller output by varying chiller fluid flow. Unfortunately this throttling forces the compressor into inefficient regions of its performance map. During variable speed operation, the water chiller vanes are maintained wide open and the Rankine cycle turbine inlet throttle valve is used to control chiller cooling capacity.

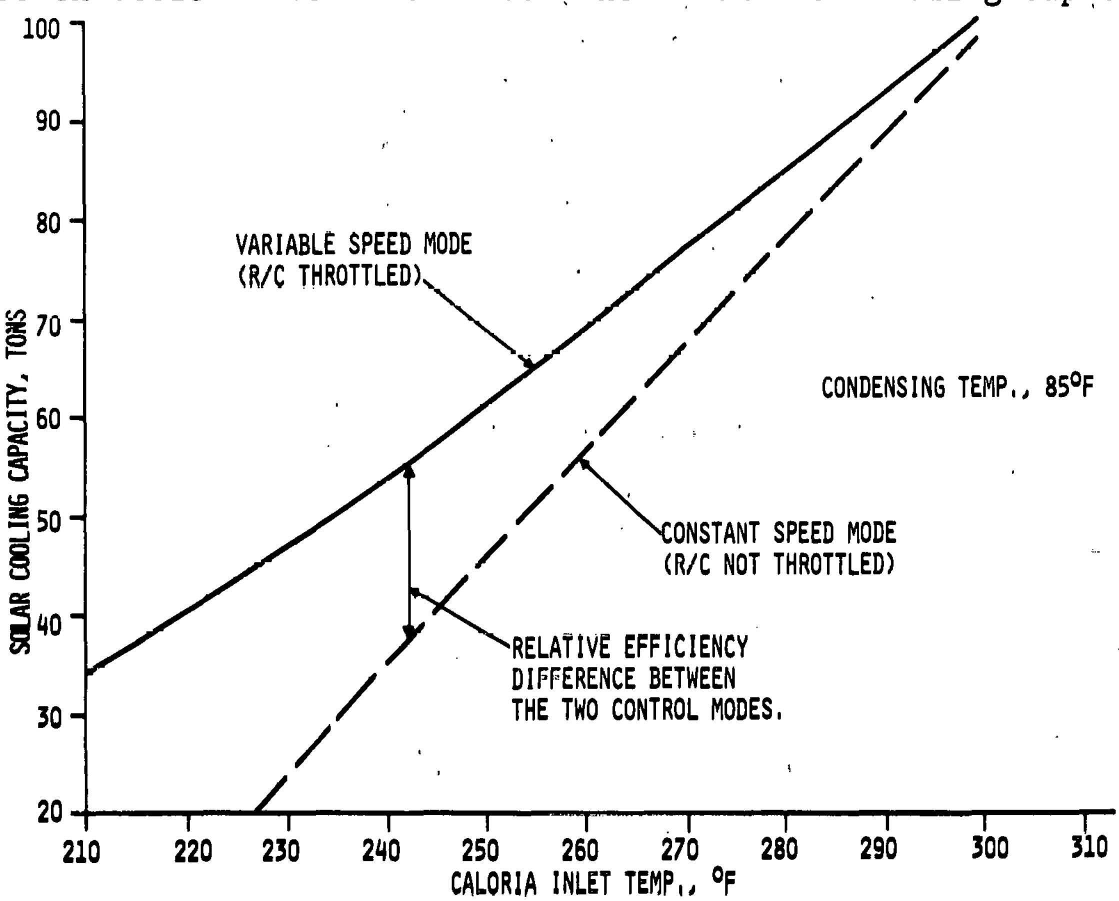

Figure 31. Performance of a Rankine Cycle Chiller

The cost of implementing variable speed control through a throttling valve is modest within the context of the entire control system, leading to the selection of the following control algorithm:

> Both units are operated by solar only (variable speed control) if the output capacity is sufficient to handle the total cooling load. If not, then one or two Rankine units are operated at constant speed, depending upon the load, with electric power providing the auxiliary energy necessary to drive the engines at full speed.

This control strategy is suboptimal in that it fails to deal with the inlet fluid temperature drop introduced by operating both units simultaneously. The development of an "optimal" algorithm will be discussed under ADVANCED CONTROL RECOMMENDATIONS.

3. Recommended Control System

The recommended control system includes weekend shutdown of ventilation and cooling systems, night thermostat setback for the heating season, economizer control, chilled water temperature reset, and a variable speed control option for the Rankine cycle chillers. This system is being implemented in the new building.

For purposes of this study it is appropriate to summarize the control components indicating the magnitude of the actuator requirements independent of any decision regarding microprocessor elements in the solar system control.

Eight pumps are associated with control of the solar loops, ranging in size from 1/12 hp to 20 hp. Additionally, there are 15 valves required for control of the solar subsystem. These include proportional valves, mixing and diverting valves and straight-through valves.

Twenty-two proportional sensors and one on/off sensor provide temperature information from a multitude of sensors: outdoor air, the solar loops, building hot water, domestic hot water, chilled water supply and return, condensor water supply and storage tanks.

State-of-the-art control implementation occurs at three levels.

- The control panel mounted in the mechanical room contains 68 relays, eight switches, seven timers, 10 differential temperature controllers, and three additional temperature controllers.

- The collector field controller consists of two differential temperature controllers, six relays and two timers.

- Each of 42 collector row local controllers includes a differential temperature controller, a motor starter, three relays and seven switches.

A detailed list of solar system components can be found in Appendix B. All of the control strategies and functions discussed earlier can and will be, implemented with this array of state-of-the-art components from Honeywell's residential commercial and industrial product line.

G. ADVANCED CONTROL RECOMMENDATIONS

It is clear that a control system using state-of-the-art components can be improved by incorporating well designed algorithms and design simplifications. In solar heating and cooling applications the benefits of "pilot production" cost reductions are already being realized in the marketing of a number of control modules. However the control cannot be optimized in the true sense of the word.

This suboptimality is particularly evident in several areas.

1. Collector Loop Control

Neither proportional control nor on/off control as currently implemented is capable of anticipation or memory, the functions which together can continually modulate setpoints to optimize collector performance.

2. System Setup and Peak Load

As shown in the tradeoff studies, there are serious peak load problems inherent in setup strategies. Thermostats on the market can be modified to introduce delays or incremental setup. However control at a sufficient level to cope with these peak load transients must continually monitor system response and make control adjustments accordingly.

3. Variable Furnace Control

The on/off furnace control that is state-of-the-art in residential heating systems is not adequate for control of an auxiliary furnace in a solar heating system. It is beyond the scope of this study

to determine how furnace performance is altered by this auxiliary role. However the fact that the furnace operates at partial load over a large portion of the heating season indicates that variable fuel and excess air control will enhance seasonal performance. State-of-the-art two-stage thermostats are not adequate for this level of furnace control.

4. Peak Load Strategies with Electric Heat Backup

In all of the residential systems for which control has been optimized, storage tank management strategies have been evaluated for avoiding peak load auxiliary heating.

The selected strategies based on outdoor temperature miss the target occasionally, though simulations attest to their effectiveness. The survey outlined more sophisticated weather forecasting strategies that involve considerable knowledge of weather in previous hours and days. State-of-the-art components do not include the memory or processing capabilities required for more sophisticated weather anticipation and response thereto.

5. Absorption Cooling Control

The development of strategies that allow an absorption unit to operate over long cycles with minimum auxiliary energy is limited with state-of-the-art components. Load prediction is based on outdoor temperatures rather than on the complex dynamics of the building and the absorption unit itself.

6. Heat Pump Control

Aspects of microprocessor control are already evidenced in heat pump controllers expected shortly on the market (see Part-I of this report). The implementation of an optimum control algorithm to minimize power consumption can certainly be enhanced when on-line computations and memory capabilities are available.

7. Rankine Cycle Cooling Control

It is evident that in a large system, Rankine cycle cooling performance can be improved by appropriately sequencing one or more units and selecting the appropriate speed control for each one.

With the capabilities inherent in microprocessor technology, optimal control techniques can lead to optimized operating algorithms to minimize power draw based on inputs from temperature and pressure sensors located at key locations in the cycle.

The year 1978 has brought developments in the marketplace that substantially alter the cost of solar control systems and overcome the limitations imposed by state-of-the-art components.

For instance any of several single-chip microcomputers described in the survey can be mounted on a printed circuit board along with a 24V power supply and interface logic to analog sensors and electromechanical relays, for a market price of $60.

This little module will be capable of providing the functions of differential temperature controllers, aquastats, reset contollers, timers and thermostat anticipators--in fact, all of the control components with the exception of the sensors and relays.

These recent developments make available to smaller buildings, in fact even to single-family residences, control functions that to date have been feasible only through advanced energy management systems for large buildings. In addition to those energy management functions cited previously, there are other advanced control features under discussion, and in some cases already available in control systems, for instance:

- Ventilation control based on the quality of the outside air.
- Control of lighting, cooling and ventilation based on building occupancy as sensed by the controller.
- Control of lighting levels based on natural light level.

Also, monitoring functions already commonplace in large energy management systems can aid immeasurably in preventative maintenance within small systems as well.

Finally, microprocessor control technology presents options as to more or less centralized control of the solar subsystem, as cited in Part I of this study. With the complexity of large commercial building systems, interface of the central building system control with the control of the various subsystems presents an interesting study in itself.

REFERENCES

1. Shipley, J.P., "Control System Design and Simulation for a Solar Cooled Building," Paper Submitted to 1975 Summer Computer Simulation Conference, San Francisco, 1975.

2. Farris, D.R. and Melsa, J.L., "Energy Savings for a Solar Heated and Cooled Building Through Adaptive Optimal Control," Paper Submitted to the IFAC VII World Congress, 1977.

3. Farris, D.R., Melsa, J.L., Murray, H.S., McDonald, T.E., and Springer, T.E., "Energy Conservation by Adaptive Control for a Solar Heated Building," presented at 1977 International Conference on Cybernetics and Society, Washington D.C., September, 1977.

4. Kovarik, M., and Lesse, P.F., "Optimal Control of Flow in Low Temperature Solar Heat Collectors", presented at the International Solar Energy Congress and Exposition, Los Angeles, July, 1975.

5. Schlesinger, R.J., "Preliminary Comparison of Proportional and Full On/Off Control Systems for Solar Energy Applications," Proceedings of the 1977 ISES Annual Meeting, June, 1977.

6. Lewis, R. Jr., "Proportional versus On/Off Differential Temperature Controls," Solar Heating and Cooling, Volume II, No. 4, August, 1977.

7. Siebers, D.L., and Viskanta, R., "Comparison of Predicted Performance of Constant Outlet Temperature and Constant Mass Flow Rate Collectors," Solar Energy, Volume 19, No. 4, pp. 411-413.

8. Eltimsahy, A.H. and Santos, E.A., Jr., "A Suboptimal Controller for a Domestic Solar Heating System Utilizing a Time Varying Price for Electricity," Proceedings of the 1977 ISES Annual Meeting, June, 1977.

9. Conway, Thomas M., "Fluid Flow Control Strategies in Flat Plate and Evacuated Tube Collectors," ASHRAE Journal, October, 1977

REFERENCES (Continued)

10. Rask, D.R., Mueller, L.J., and Pejsa, J.H., "Low Cost Air Heater Final Report," prepared for the U.S. Energy Research and Development Administration, August, 1977.

11. Nelson, L.W., "Predicting Control Performance of Residential Heating Systems with an Analog Computer," IEEE Transactions on Industry Applications, Volume 1A-10, No. 6, November/December, 1974.

12. Tobias, J.R., and Nelson, L.W., "Energy Savings in Residential Building," ASHRAE Journal, February, 1974.

13. Nelson, L.W., and Tobias, J.R., "Studies of Control Applications for Energy Conservation," ASHRAE Symposium, Los Angeles, Summer, 1974.

14. "Identifying Small Building Types and Requirements for Microprocessor Energy Management Systems," (Task Orders 1 and 2), Final Report CR-04-60882, JRB Associates, March, 1977.

15. Shavit, G., "Energy Conservation and Fan Systems: Computer Control with Floating Space Temperature," ASHRAE Journal, October, 1977.

16. Ward, D.S., Smith, C.C., and Ward, J.C., "Operational Modes of Solar Heating and Cooling Systems," Solar Energy Volume 19, No. 1.

17. Wahlig, M., "Control System for Combined Solar Heating and Cooling Systems," Lawrence Berkeley Laboratories; Pacific Gas and Electric Company, San Francisco, California; January 31, 1975.

18. Van't Land, J.A., Woods, J.F., and Peterson, P.W., "Comparative Performances of Solar Assisted Residential Heat Pump Systems in Northern Climates," Energy Research Institute, Iowa State University, March, 1978.

19. Andrews, J.W., Kush, E.A., and Metz, P.D., "A Solar Assisted Heat Pump System for Cost Effective Space Heating and Cooling," Brookhaven National Laboratory, DOE Contract No. EY-76-C-02-0016.

REFERENCES (Continued)

20. Mitchell, J.W., Freeman, T.L., and Beckman, W.A., "Combined Solar Heat Pump Systems for Residential Heating," University of Wisconsin-Madison, for publication in Solar Progress.

21. Zabinski, M.P., and Parlange, J.Y., "Thermostat Down," ASHRAE Journal, January, 1977.

22. Nelson, Lorne W., "Reducing Fuel Consumption with Night Setback," ASHRAE Journal, August, 1973.

23. Pilati, D.A., "Energy Conservation Potential of Winter Thermostat Reductions and Night Setback," Oak Ridge National Laboratory, Tennessee, February, 1975.

24. Nelson, L.W., and MacArthur, W., "Energy Savings Through Thermostat Setback," copyright Honeywell Inc., 1977.

25. Nelson, L.W., et. al., "The Analog Computer as a Product Design Tool," ASHRAE Journal, November 1965.

26. Tobias, J.R., McConnell, R.R., and Nelson, L.W., "Reducing Energy Consumption During the Cooling Season," ASHRAE Journal, June, 1976.

27. Newton, A.B., "Making Solar Heating and Cooling Practical," Sydney, Australia, June, 1977.

28. Energy Marketing, "Office Uses Off Peak Thermal Storage," Electrical World, November 1, 1975.

29 Hughes, P.J., Beckman, W.A., and Duffie, J.A., "Simulation Study of Several Solar Heating Systems with Off Peak Auxiliary," ISES Annual Meeting, Winnipeg, August, 1976, Volume IV.

30. Hughes, P.J., Freeman, T.L., and Grunes, H., "Design of an Experimental Solar House Heating System," American Society of Agricultural Engineers, 1976 Winter Meeting, Chicago, December, 1976.

31. Corpening, S.L., Reppen, N.D., and Ringlee, R.J., "Experience with Weather Sensitive Load Models for Short and Long Term Forecasting," IEEE Power Engineering Society Winter Meeting, New York, January, 1973.

REFERENCES (Continued)

32. Proposal to ERDA: "Control Strategies for Peak Load Reduction," Honeywell Energy Resources Center, Minneapolis, June, 1977.

33. "Energy Saving Thermostats," Consumer Reports, October, 1977.

34. Woods, J.E., and Peterson, P.W., "Evaluation of an Energy Conserving Research House Involving Multi-Model Operation of Solar and Heat Pump Systems," ISES Annual Meeting, Orlando, June, 1977.

35. Energy and Environment Division, Annual Report 1976, Lawrence Berkeley Laboratories-University of California.

36. Technical Support Package for Tech Brief LAR-12134, NASA Technology Utilization House, Langley Research Center, Winter, 1976.

37. Cook, Jerry, "How to Specify Pumps for Solar Systems," Solar Engineering, May, 1977.

38. Hobmar Homes, Inc., Honeywell Inc., etc., "Solar Active/Passive Air Heating Demonstration," Grant No. H-8140, U.S. Department of Housing and Urban Development, 1977.

39. Bonne, U., Janssen, J.E., Johnson, A.E., and Wood, W.T., "Residential Heating Equipment HFLAME Evaluation of Target Improvements," Final Report to Center for Building Technology, NBS, Contract No. T62709, September, 1976.

40. Murray, H.S., Trump, M.A., and Hedstrom, J.C., "Solar Heating and Cooling System Performance Experiments in the Los Alamos National Security and Resources Study Center," Proceeding of the 1977 ISES Annual Meeting, June, 1977.

41. Knoll, R.H., Miao, D., Hamlet, I.L., and Jensen, R.N., "Initial Operation of a Solar Heating and Cooling System in a Full Scale Solar Building Test Facility," NASA Technical Memorandum TM X-73519, August, 1976.

REFERENCES (Concluded)

42. "Residential Water Heating: Fuel Conservation, Economics, and public Policy," prepared for the National Science Foundation, Rand Corporation, R-1498-NSF, May, 1974, James J. Mutch.

43. "Mixed Strategies for Energy Conservation and Alternative Energy Utilization in Buildings," Honeywell Inc., Contract for ERDA No. E (04-3)- 1234.

44. Beckman, Wm. A., "Problems of Solar Cooling Systems," Sunworld, November 1977, pp. 2-6.

45. Ward, D.S., Lof, G.O.G., and Useki, T., "Cooling Subsystem Design in CSU Solar House III," Solar Energy, Volume 20, Number 2, 1978, pp. 119-126.

46. "Guide to Room Thermostats," Appliance Manufacturer, September, 1971, pp. 89-91, prepared by Honeywell.

APPENDICES

APPENDIX A
ECONOMIC EVAULATION GUIDELINES

Inasmuch as the work undertaken herein is directed toward cost-effective control system design, the bases for the economic evaluations will first be defined.

One index frequently used is that of <u>payback period</u> -- the length of time it takes the total net energy savings to equal the initial investment. The calculation of payback period may well neglect recurring costs and benefits, but has the advantage of providing a ready tool for choosing among alternatives. It is being applied in our evaluations for selection among alternate strategies and control configurations within each system and application. One strategy is selected over another based on the payback advantage. The relatively simple algorithm is derived as follows. If N is the payback period in years, then

$$\Delta I = \Delta E_o \sum_{J=1}^{N} \left(\frac{1+\varepsilon}{1+i}\right)^j - \Delta P_o \sum_{j=1}^{N} \left(\frac{1+\varepsilon}{1+i}\right)^j$$

Here ΔI represents the initial hardware cost of implementing Strategy A over some baseline control design, ΔE_o the additional annual fuel energy savings and ΔP_o any additional operating costs, all at current rates. The projected annual escalation rate is denoted by ε and the current discount rate by i. If in the short term the fuel and power escalation rates are not too different, a closed-form solution to equation (1) can be derived:

$$N = \log \left[\frac{\Delta I}{\Delta E_o - \Delta P_o} \cdot \frac{r - 1}{r} + 1 \right] / \log r$$

where, $r = \dfrac{1 + \varepsilon}{1 + i}$.

E_o = Annual fuel energy savings, at 1977 rates (Table A-1),

I = Installed costs, system plus control,

P_o = Annual power operating costs, at 1977 rates (Table A-1),

i = Money discount rate,

ε = Escalation rate for fuel and power energy (Table A-2) and for maintenance (7 percent by FEA inflation projections).

In these analyses, the energy costs, the rate of energy cost escalation and the market cash discount rate are all key parameters. Energy rates vary by region and by fuel, as well as between residential and commercial applications. These variations do indeed result in control recommendations that differ according to the backup energy source, as well as by region and application. For purposes of the evaluations herein the Department of Energy cost predictions will be applied.* Figures A-1 through A-4 present the 1977 cost projections by region, for both electricity and natural gas, the selected energy sources in this study. In Table A-1 are tabulated energy costs for 1977 (E_o) and in Table A-2 the projected average annual escalation rates (ε), as calculated from the FEA data. A value of 2 percent has been selected for the real discount rate corresponding to a current market discount rate of about 10 percent.*

*FEA Energy Price Regulations, Federal Register, Vol. 42, No. 125 June 29, 1977.

Table A-1. 1977 Energy Costs in 1977 Dollars

FEA REGION	SELECTED CITY	RESIDENTIAL GAS ($/1000 FT^3)	RESIDENTIAL ELECTRICITY ($/kWH)	COMMERCIAL GAS ($/1000 FT^3)	COMMERCIAL ELECTRICITY ($/kWH)
MIDWEST	MINNEAPOLIS	$1.97	$0.03686	$1.68	$0.03622
CENTRAL	OMAHA	$1.57	$0.03605	$1.21	$0.03344
NEW/YORK NEW JERSEY	NEW YORK	$3.02	$0.05453	$2.46	$0.05591
SOUTH ATLANTIC	ATLANTA	$1.94	$0.03259	$1.53	$0.03385
SOUTHWEST	ALBUQUERQUE	$1.66	$0.03657	$1.24	$0.03318

FEA RULES & REGULATIONS, JUNE 1977.

Table A-2. Projected Average Annual Energy Escalation Rates Through 1990 Over and Above Inflation Increases

FEA REGION	SELECTED CITY	RESIDENTIAL GAS	RESIDENTIAL ELECTRICITY	COMMERCIAL GAS	COMMERCIAL ELECTRICITY
MIDWEST	MINNEAPOLIS	5.1%	0.8%	5.6%	1.0%
CENTRAL	OMAHA	4.8%	0.3%	9.1%	1.2%
NEW YORK/ NEW JERSEY	NEW YORK	3.2%	-0.5%	3.8%	0.4%
SOUTH ATLANTIC	ATLANTA	4.8%	0.9%	5.5%	0.5%
SOUTHWEST	ALBUQUERQUE	4.4%	1.9%	8.4%	2.5%

FEA RULES & REGULATIONS, JUNE 1977

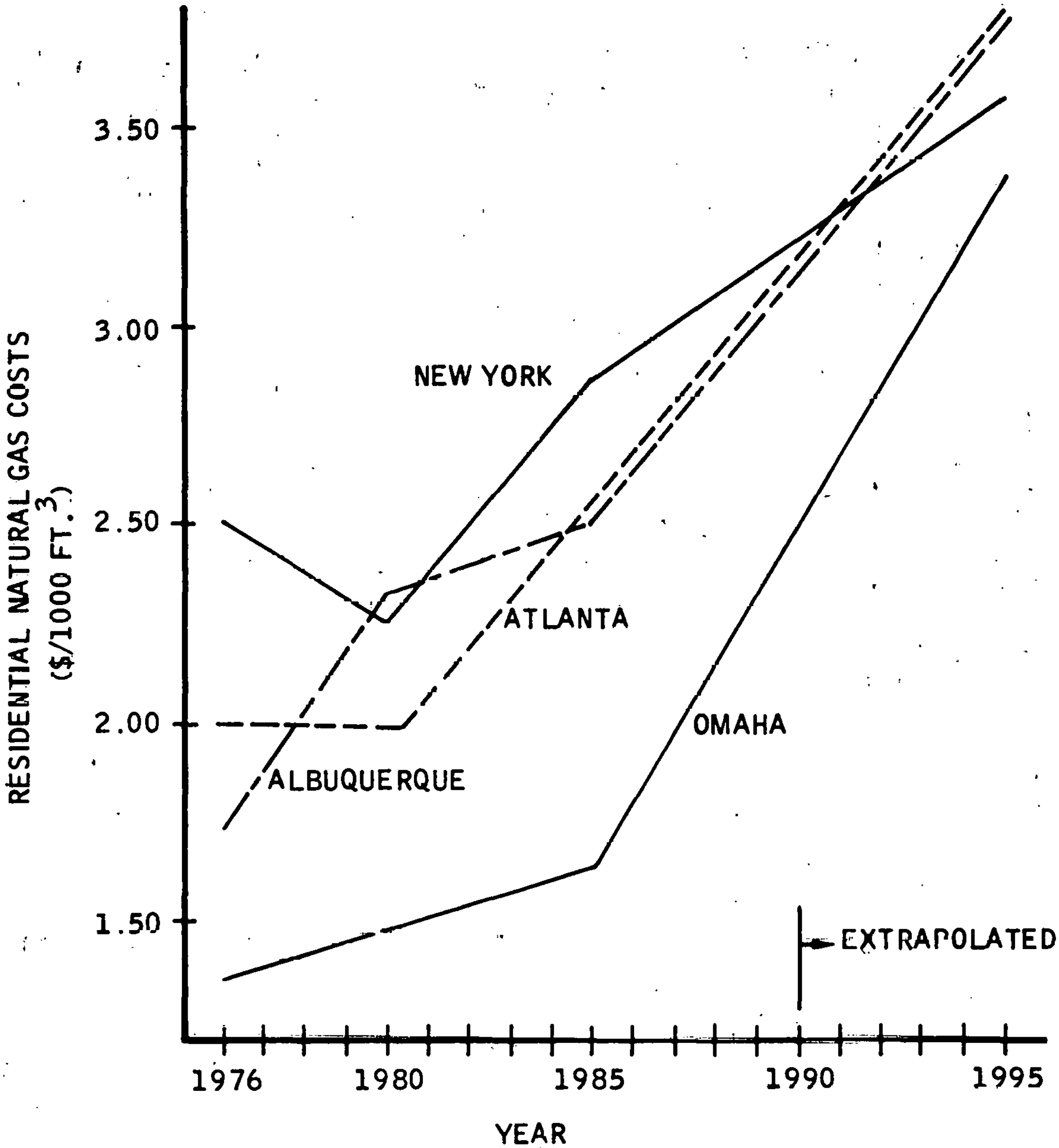

Figure A-1. Source: FEA 12/30/76; 1975 Dollars

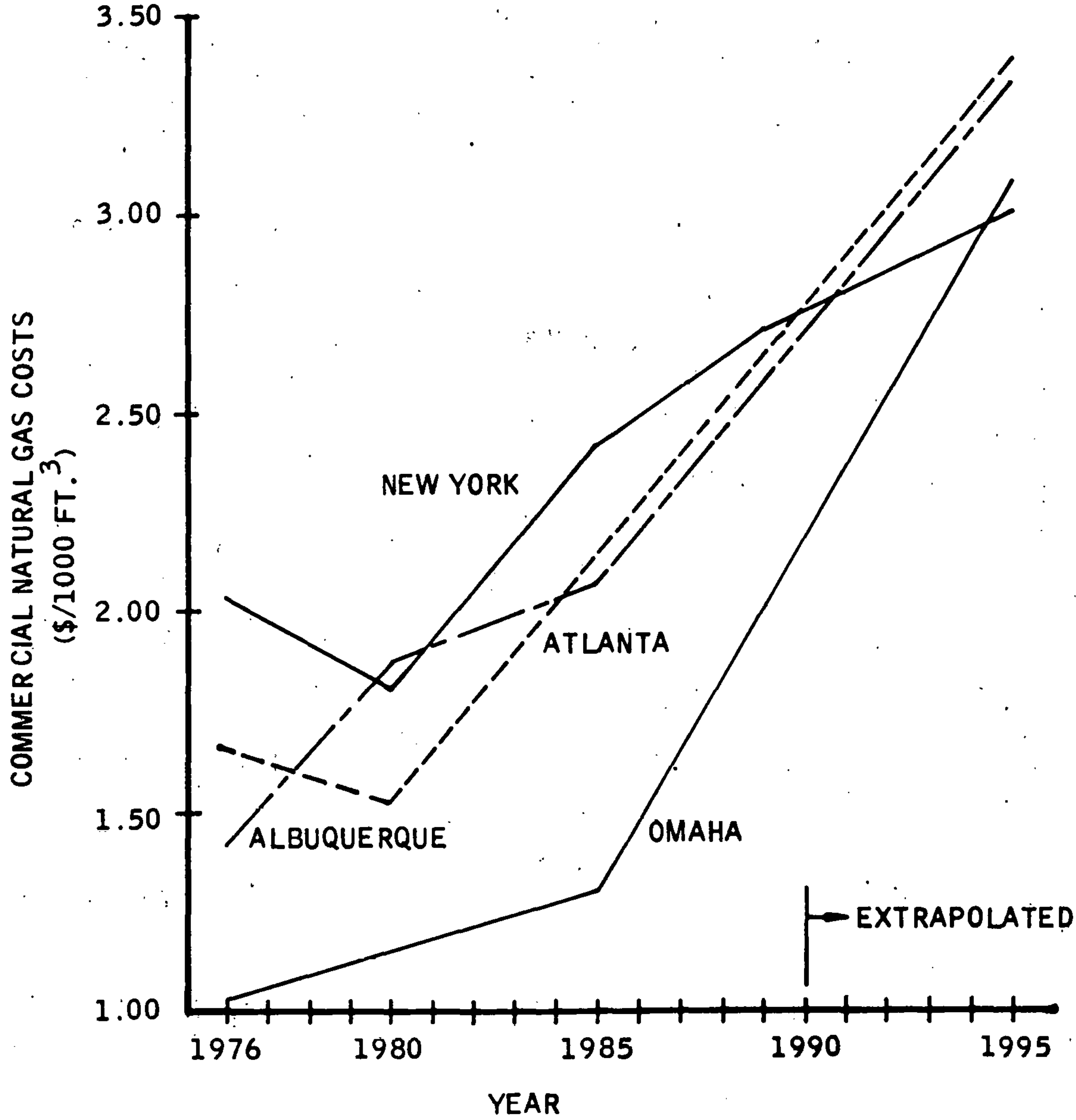

Figure A-2. Source: FEA 12/30/76; 1975 Dollars

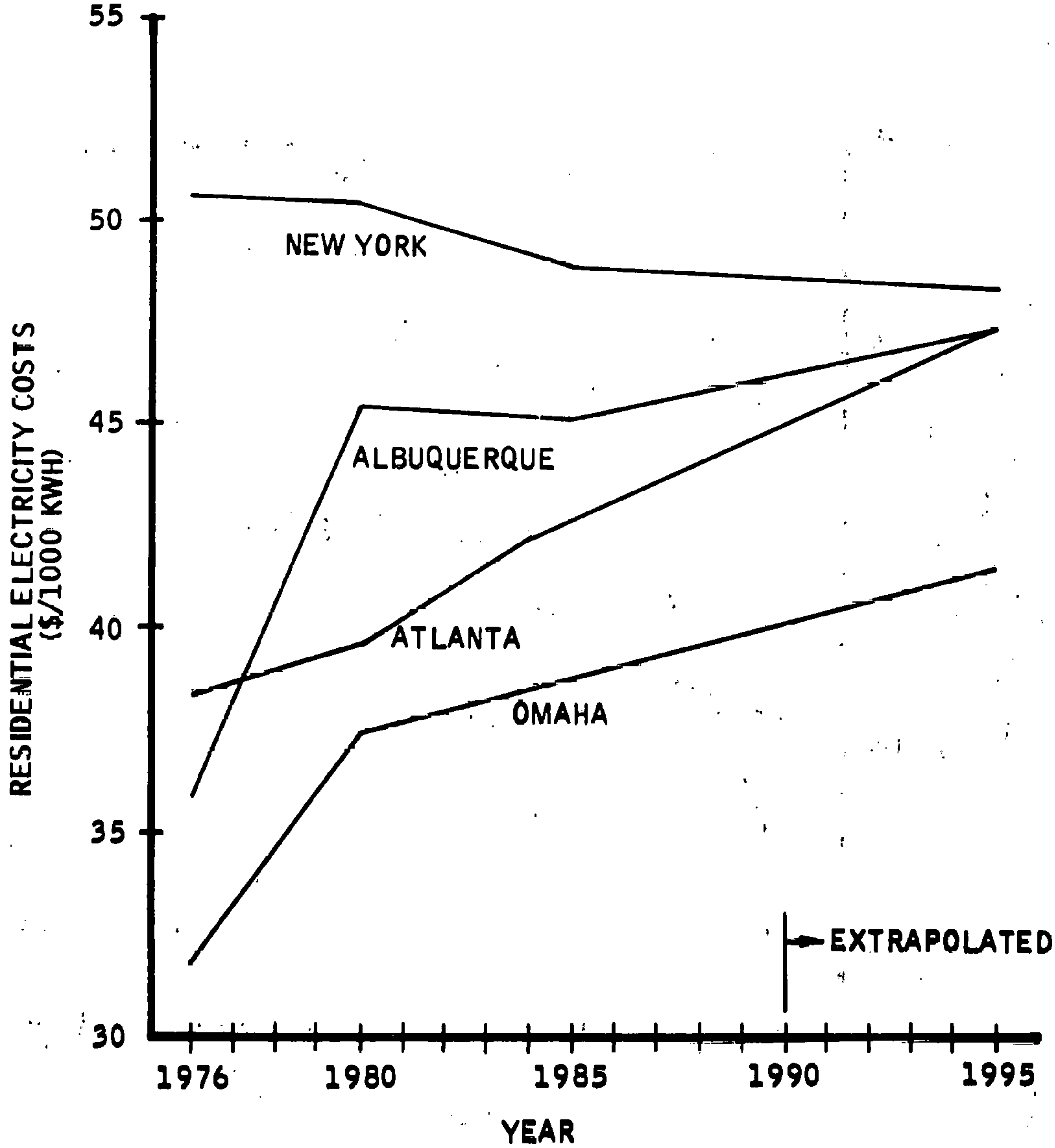

Figure A-3. Source: FEA 12/30/76; 1975 Dollars

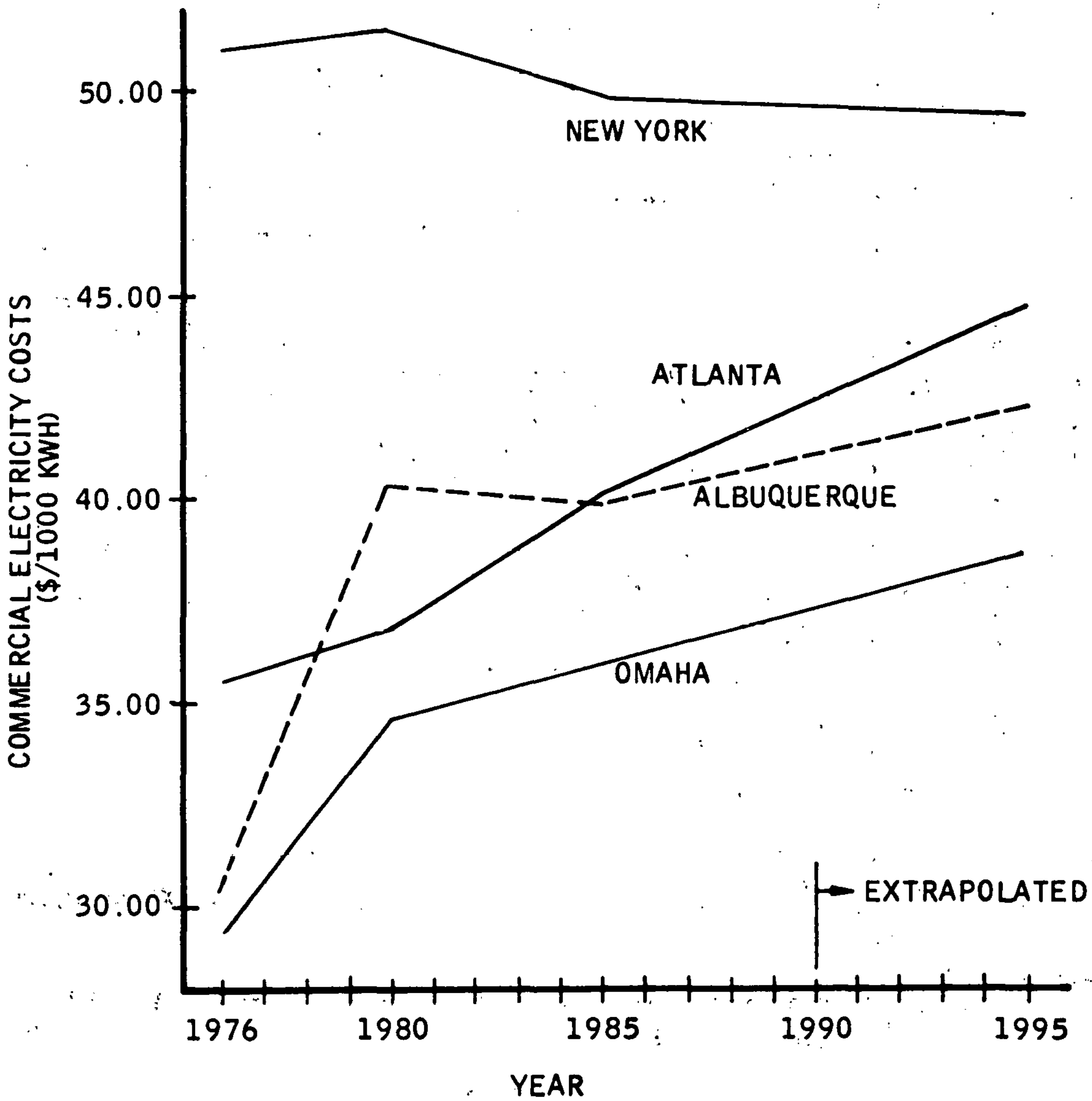

Figure A-4. Source: FEA 12/30/76; 1975 Dollars

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

APPENDIX B

CONTROL SYSTEM COMPONENTS

Table B-1. Control Components for a Single-Family Residential Hydronic Solar Heating System

COMPONENT	PART NUMBER	CONSUMER COST (1977)	BASELINE SYSTEM	OPTIMIZED SYSTEM
THERMOSTAT	HW T872-D4011	$ 68.18	1	
SUBBASE	HW Q672 E1001	14.46	1	1
MODIFIED FOR AUTO. RESET	HW T872 D1300	130.00		1
SOLAR CONTROL BOX & LABOR		201.82	1	1
RELAYS	HW R8222 N1011	14.35	1	
	HW R8222 F1011	9.00	1	
	HW R8222 Y1007	32.20	1	
	HW R4222 F1000	9.36	1	
	HW R8230 G1009	13.75		1
	HW R8225 A1017	20.88		2
AQUASTATS	HW L4008 D1013	42.02	1	
	HW L4008 D1050	41.99	1	
	HW L6008 C1206	42.44		1
DIFFERENTIAL TEMP. CONTROLLER	HW R7412 A1004	112.00	1	1
SENSORS	HW C773 A1006	14.04	2	2
	HW C773 B1005	25.96		1
TRANSFORMER	HW AT88 A1005	32.20		1
ECONOMIZER OPTION				
ENTHALPY CONTROLLER	HW H205 A1012	63.13 + 6.50*		1
OFFPEAK & COOLING OPTION				
RESET CONTROLLER	HW T475 A1016	112.17		1
RELAYS	HW R8225 A1017	20.88		1
	HW R8246 A1017	37.39		1
INTERMATIC TIMER	HW V41075	84.00		1
ACTUATORS				
VALVES	HW V8044 A1044	107.04 + 30.80*	2	2
	HW V4044 A1191	55.25 + 15.40*	1	
BELL & GOSSETT PUMPS	B & G 60-11	180.00 + 37.00*	1	2
	B & G PR	200.00 + 37.00*	2	
ECONOMIZER OPTION				
DAMPER	HW D640 A9009	48.00 + 5.00*		3
MOTOR	HW M836	112.17 + 13.00		1

*LABOR COST ESTIMATES FROM BUILDING CONSTRUCTION COST DATA 1977, 35TH ANNUAL EDITION, MEANS COMPANY.

Table B-2. Control Components for a Single-Family Residential Active/Passive Solar Air Heating System

COMPONENT	PART NUMBER	CONSUMER COST (1977)	BASELINE SYSTEM	OPTIMIZED SYSTEM
THERMOSTAT	HW T872 D1011	$ 68.18	1	
SUBBASE	HW Q672 E1050	14.46	1	1
MODIFIED FOR AUTO. RESET	HW T872 D1300	130.00		1
SOLAR CONTROL BOX & LABOR		201.82	1	1
RELAYS	HW R8230 C1024	13.75	2	
	HW R4230 C1049	13.75	2	
	HW R4230 F1000/1	12.24	2	
	HW R8222 D1014	16.53	1	
	HW R8225 A1017	20.88		1
AQUASTATS	HW L6008 C1065	42.44	1	
	HW L4008 B1013	42.02	1	
	HW L4008 C1037	41.99		1
DIFFERENTIAL TEMP. CONTROLLER	HW R7412 A1012/04	112.00	2	2
SENSORS	HW C773 A1006	14.04	3	2
	HW C773 B1005	25.96	1	1
ECONOMIZER OPTION				
ENTHALPY CONTROLLER	HW H205 A1012	63.13 + 6.58*	1	1
OFFPEAK & COOLING OPTION				
RESET CONTROLLER	HW T475 A1016	112.17		1
RELAYS	HW R8225 A1017	20.88		1
	HW R8246 A1004	37.39		1
INTERMATIC TIMER	HW V41075 CR	84.00		1
ACTUATORS				
DAMPERS	HW D640 A9009	48.00 + 5.10*	4	5
	HW D640 A9157	87.00 + 5.10*	1	
	HW D640 24" x 8"	87.00 + 5.10*	1	
MOTORS	HW M836 B1033/ A1042	112.17 + 13.00*	3	2
	HW M945	335.66 + 13.00*	1	
TRANSFORMERS	HW AT72 D1683	10.22 + 3.25*	4	
	HW AT88 A1005	32.20 + 3.25*	1	
ECONOMIZER OPTION				
DAMPERS	HW 0640 A9009	48.00 + 5.10*		3
MOTOR	HW M836 A1042	112.17 + 13.00*		1

*LABOR COST ESTIMATES FROM BUILDING CONSTRUCTION COST DATA 1977 35TH ANNUAL EDITION, MEANS COMPANY

Table B-3. Control Components for a Single-Family Residential Hydronic Solar Heating/ Cooling System

COMPONENT	HONEYWELL PART NUMBER	NATURAL GAS BACKUP AUX. BOILER	NATURAL GAS BACKUP AUX. FURNACE	ELECTRIC BACKUP
THERMOSTAT, TWO-STAGE	T872 D1003	1	1	1
SUBBASE	Q672 B1079	1	1	1
TRANSFORMER	AT88 A1005	1	1	1
RELAYS	R8230 G1009	5	5	5
	R8230 C1009	1	1	1
DIFFERENTIAL TEMP. CONTROLLER	R7412 ADVANCED	1	1	1
ENTHALPY CONTROLLER	H205 A1012			
OUTDOOR RESET				
CONTROLLER	T475 XXXXX	2	1	2
AQUASTAT	L6008 A1010	1	2	1
SENSORS	C773 A1006	2	2	2
DUAL	C773 B1005	1	1	1
VALVES, DIVERTING	Y534 A1007	4	4	4
	Y534 A1015	1	-	1
DAMPERS	D640	3	3	3
MOTOR	M836 A1042	1	1	1
LINKAGE	Q298 B1065	2	2	2

Table B-4. Control Components for a Large Commercial Building Solar Concentrating Collector Heating/Cooling System

PUMPS IN THE SOLAR LOOP	
PUMP	HORSEPOWER
DOMESTIC WATER SYSTEM OIL CIRCULATING PUMP	1/3
PRIMARY LOOP START-UP PUMP	15
SECONDARY LOOP CIRCULATING PUMP	20
HEATING SUBSYSTEM OIL CIRCULATION PUMP	5
PRIMARY LOOP CIRCULATION PUMP	20
DOMESTIC WATER SYSTEM WATER CIRCULATION PUMP	1/12
RANKINE TURBINE #1 CONDENSER WATER PUMP	10
RANKINE TURBINE #2 CONDENSER WATER PUMP	10

Table B-4. Control Components for a Large Commercial Building Solar Concentrating Collector Heating/Cooling System

LOOP	HONEYWELL VALVE NO.	TYPE	QUANTITY
DOMESTIC HOT WATER	1605	3-WAY MIXING	1
SECONDARY SOLAR LOOP STORAGE	1901	2-POSITION DIVERTING	1
SECONDARY SOLAR LOOP TEMP.	1601	3-WAY MIXING	1
SECONDARY SOLAR LOOP BALANCING	9101	2-WAY STRAIGHT-THRU, NO	2
R/C UNIT #1	9131	2-WAY STRAIGHT-THRU, NC	1
R/C UNIT #2	9131	2-WAY STRAIGHT-THUR, NC	1
BUILDING HEAT CONVERTER HT2	1601	3-WAY MIXING	1
NORTH & EAST RADIATION	VP516	3-WAY MIXING	1
NORTH & EAST RADIATION	VP514	PROPORTIONAL	1
SOUTH & WEST RADIATION	VP516	3-WAY MIXING	1
SOUTH & WEST RADIATION	VP514	PROPORTIONAL	1
TANK 1 SHUTOFF	9101	2-WAY STRAIGHT-THRU, NO	1
TANK 2 SHUTOFF	9101	2-WAY STRAIGHT-THRU, NO	1
SOLAR HEAT EXCHANGER BYPASS	9101	2-POSITION DIVERTING	1
		TOTAL:	15

Table B-4. Control Components for a Large Commercial Building Solar Concentrating Collector Heating/Cooling System (Continued).

LOOP	DESCRIPTION	TYPE	QUANTITY	HONEYWELL CATALOG NUMBER
PRIMARY SOLAR LOOP	200 Ω PT. DUAL ELEMENT	PROPORTIONAL	2	WSPIC2-9-3A-RW3-D
TANK SENSOR	100 Ω PT. THREE ELEMENT	"	2	WSPOB1-18 1/2-3A
TANK SENSOR	100 Ω PT. DUAL ELEMENT	"	6	WSPOB1-18 1/2-3A
SECONDARY SOLAR LOOP	200 Ω PT. ELEMENT	"	1	WSPIC2-9-3A-RW3
CHILLED WATER RETURN	200 Ω ONE ELEMENT	"	1	WSPIC2-9-3A-RL2
CHILLED WATER RETURN	200 Ω TWO ELEMENT	"	1	WSPIC2-9-3A-RL2-D
CHILLED WATER SUPPLY	200 Ω ONE ELEMENT	"	1	WSPIC2-9-3A-RL2
BUILDING HEATING WATER RETURN	200 Ω TWO ELEMENT	"	1	WSPIC2-9-3A-RL2-D
BUILDING HEATING WATER RETURN	200 Ω ONE ELEMENT	"	1	WSPIC2-9-3A-RL2
CONDENSOR WATER SUPPLY	200 Ω TWO ELEMENT	"	1	WSPIC2-9-3A-RL2-D
CHILLED WATER SUPPLY	PNEUMATIC TEMPERATURE SENSOR	"	1	LP914 A1060
NORTH & EAST RADIATION ZONE TEMPERATURE	PNEUMATIC TEMPERATURE SENSOR	"	1	LP914 A1052
SOUTH & WEST RADIATION ZONE TEMPERATURE	PNEUMATIC TEMPERATURE SENSOR	"	1	LP914 A1052
DOMESTIC HOT WATER SUPPLY	PNEUMATIC TEMPERATURE SENSOR	ON/OFF	1	LP914 A1052
OUTSIDE AIR TEMPERATURE SENSOR EAST	PNEUMATIC TEMPERATURE SENSOR	PROPORTIONAL	1	LP914 A1011
OUTSIDE AIR TEMPERATURE SENSOR WEST	PNEUMATIC TEMPERATURE SENSOR	"	1	LP914 A1011
TOTAL: 22 Proportional; 1 On/Off				

Table B-4. Control Components for a Large Commercial Building Solar Concentrating Collector Heating/Cooling System (Continued)

RELAYS	HONEYWELL CATALOG NO.	QUANTITY
ELECTRIC	R4222 D1005	35
	R4214 E1002	2
	R4222 D1065	1
	14500 575-002	1
	14501 295-001	1
	R7350 H1037	1
	R7350 B1014	2
	R7082 C1041	10
ELECTRONIC-PNEUMATIC	RP417 B1007	11
PNEUMATIC	RP908 B1037	3
	RP908 A1021	1
SWITCHES		
ELECTRIC SWITCHES	4 TL 1-3	1
	11TS115-3	3
	4 1 P8205	1
PNEUMATIC SWITCHES	P643 A1007	3
TIMERS		
	157520682	2
	804132 DAA	1
	804132 NAA	1
	804132 PAA	2
	804132 LAA	1

Table B-4. Control Components for a Large Commercial Building Solar Concentrating Collector Heating/Cooling System (Concluded)

DESCRIPTION	HONEYWELL CATALOG NO.	QUANTITY
MISCELLANEOUS CONTROLS		
SECONDARY SOLAR LOOP INDICATOR & CONTROL	T654 A1602	3
CONDENSOR WATER TOWER FAN	T678	1
PNEUMATIC TEMPERATURE CONTROLLER SECONDARY SOLAR LOOP	TP954 A1475	2
PNEUMATIC TEMPERATURE CONTROLLER SOLAR SUPPLY TEMPERATURE TO BUILDING HEATER #2	TP954 A1475	1
PNEUMATIC TEMPERATURE CONTROLLER SOLAR SUPPLY TEMPERATURE TO DHW HEATER	TP954 A1475	1
PANEL MOUNTED CONTROLS		
DIFFERENTIAL TEMPERATURE CONTROLLERS	R7082 C1041	10
TEMPERATURE CONTROLLERS	R7350 H1037 R7350 B1014	1 2
COLLECTOR FIELD CONTROLLER		
DIFFERENTIAL TEMPERATURE CONTROLLER	R7406	2
RELAY	CS1A-120000-U-24DC	6
TIMER	RZU2-51-4H-24DC	2
COLLECTOR ROW LOCAL CONTROLLER		
DIFFERENTIAL TEMPERATURE CONTROLLER	R7412	1
MOTOR STARTER	CAUT1-10E1-M-120AC-24A	1
RELAYS	CS1A-120000-U-24DC	3
SWITCHES	NO PART NUMBER	7

THIS PAGE

WAS INTENTIONALLY

LEFT BLANK

APPENDIX C

SIMULATION RESULTS FOR FOUR REGIONS HYDRONIC SOLAR HEATING/COOLING SYSTEMS

Table C-1. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Omaha (1953 Weather)

TRADEOFFS	SOLAR PERFORMANCE HEATING AND CHARGING STORAGE				AUXILIARY SPACE HEAT (60% EFFICIENCY FOR GAS)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY FOR GAS)		FURNACE BLOWER OPERATION (HEATING AND COOLING)		*TOTAL HEATING COSTS $/YEAR
	SPACE HEATING BTUS/YEAR	DOMESTIC HOT WATER BTUS/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	AUX. OUTPUT BTUS/YEAR	$/YEAR GAS (ELECTRIC)	AUX. OUTPUT BTUS/YEAR	$/YEAR GAS (ELECTRIC)	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM	32.0×10^6	12.2×10^6	1112	$44.48	42.7×10^6	$111.73 ($500.02)	7.4×10^6	$14.52 ($86.65)	1245	$49.80	$220.53 ($680.95)
WITH 8-HOUR -5°F NIGHT SETBACK	30.1×10^6	12.4×10^6	1081	$43.24	39.0×10^6	$102.05 ($456.69)	7.2×10^6	$14.13 ($84.31)	1156	$46.24	$205.66 ($630.48)
WITH SETBACK PROPORTIONAL CONTROL OF ABSORPTION UNIT	30.4×10^6	12.6×10^6	839	$33.56	38.7×10^6	$101.26 ($453.18)	7.0×10^6	$13.74 ($81.97)	1659	$66.36	$214.92 ($637.34)
WITH SETBACK ON/OFF CONTROL OF ABSORPTION UNIT NO AUX. COOLING	30.1×10^6	12.4×10^6	872	$34.88	39.0×10^6	$102.05 ($456.69)	7.2×10^6	$14.13 ($84.31)	1165	$46.60	$197.66 ($622.48)
WITH SETBACK DIFFERENTIAL TEMP. CONTROL OF STORAGE NO AUX. COOLING	31.4×10^6	12.1×10^6	1145	$45.80	37.7×10^6	$98.65 ($441.47)	7.5×10^6	$14.72 ($87.83)	1189	$47.56	$206.73 ($622.66)

FEA 1977 NATURAL GAS RATES FOR CENTRAL REGION: $1.57/MILLION BTUS.
FEA 1977 ELECTRIC RATES FOR CENTRAL REGION: $.04/KWH OR $11.71/MILLION BTUS.

PARASITIC POWER REQUIREMENTS
- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS
- AIRHANDLER BLOWER (1/3 HP) 459 WATTS
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-2. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Omaha (1953 Weather)

TRADEOFFS	SOLAR PERFORMANCE COOLING ONLY				AUXILIARY BOILER ENERGY (60% EFFICIENCY FOR GAS)		ECONOMIZER COOLING BTUS/YEAR (HOURS)	COOLING LOAD NOT SATISFIED BTUS/YEAR (HOURS)	TOTAL COOLING COSTS $/YEAR	TOTAL ENERGY COSTS $/YEAR
	SPACE COOLING BTUS/YEAR	SOLAR SYSTEM PUMPS KWHS/YEAR	ABSORPTION SYSTEM PUMPS KWHS/YEAR	ELECTRIC COST $/YEAR	AUX. OUTPUT BTUS/YEAR (HOURS)	$/YEAR (ELECTRIC) GAS				
SETBACK SYSTEM ON/OFF CONTROLS OF ABSORPTION UNIT BOOST TO 170°F WITH AUXILIARY BOILER	16.9×10^6 x 0.68 (COP)	187	251	$17.52	2.5×10^6 (132)	$6.55 ($29.28)	1.6×10^6 (851)	-	$24.07 ($46.80	$229.73 ($677.28)
PROPORTIONAL CONTROL OF ABSORPTION UNIT BOOST TO 170°F WITH AUXILIARY BOILER	12.9×10^6 x 0.74 (COP)	515	345	$34.40	2.8×10^6 (244)	$7.33 ($32.79)	1.6×10^6 (852)	-	$41.73 ($67.19)	$256.65 ($702.53)
ON/OFF CONTROL OF ABSORPTION UNIT NO AUXILIARY	15.3×10^6 x 0.68 (COP)	156	193	$13.96	-	-	1.6×10^6 (851)	1.1×10^6 (135)	$13.96	$211.62 ($636.36)
DIFF. TEMP. CONTROL ON/OFF CONTROL OF ABSORPTION UNIT NO AUXILIARY UNIT	15.3×10^6 x 0.68 (COP)	156	193	$13.96	-	-	1.6×10^6 (850)	1.1×10^6 (135)	$13.96	$220.69 ($636.62)

COP IS SEASONAL COEFFICIENT OF PERFORMANCE.

FEA 1977 NATURAL GAS RATES FOR CENTRAL REGION: $1.57/MILLION BTUS.

FEA 1977 ELECTRIC RATES FOR CENTRAL REGION $.04/KWH OR $11.71/MILLION BTUS.

PARASITIC POWER REQUIREMENTS

- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS.
- AIR HANDLER BLOWER (1/3 HP) 459 WATTS
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-3. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Atlanta (Nashville 1955 Weather)

	SOLAR PERFORMANCE HEATING AND CHARGING STORAGE				AUXILIARY SPACE HEAT (60% EFFICIENCY FOR GAS)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY FOR GAS)		FURNACE BLOWER OPERATION (HEATING AND COOLING		TOTAL HEATING COSTS $/YEAR
	SPACE HEATING BTUS/YEAR	DOMESTIC HOT WATER BTUS/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	24.4×10^6	11.4×10^6	900	$29.33	25.1×10^6	$81.05 ($239.38)	5.7×10^6	$13.90 ($54.72)	1104	$35.98	$160.26 ($359.41)
NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	22.5×10^6	11.6×10^6	921	$30.02	22.3×10^6	$72.12 ($212.99)	5.5×10^6	$13.39 ($52.74)	1016	$33.11	$148.64 ($328.86)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	23.6×10^6	11.4×10^6	912	$29.74	21.2×10^6	$68.55 ($202.47)	5.8×10^6	$13.98 ($55.93)	1071	$34.91	$147.18 ($322.15)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK NO AUXILIARY COOLING	23.3×10^6	11.2×10^6	978	$31.89	21.5×10^6	$69.60 ($205.57)	5.9×10^6	$14.37 ($56.60)	928	$30.24	$146.10 ($324.30)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F + $(T_{AMB} - 78)/2$.	23.6×10^6	11.4×10^6	931	$30.35	21.2×10^6	$68.50 ($202.32)	5.7×10^6	$13.92 ($54.84)	1027	$33.47	$146.34 ($320.98)
SELECTED SYSTEM INLET TO HOUSE SOLAR COIL BOOSTED TO 125°F - .8676 T_{AMB}	22.3×10^6	11.9×10^6	1126	$36.71	23.4×10^6	$75.74 ($223.72)	5.2×10^6	$12.67 ($49.91)	1156	$37.68	$162.80 ($348.02)

FEA 1977 NATURAL GAS RATES FOR SOUTH ATLANTIC REGION: $1.94/MILLION BTUS.
FEA 1977 ELECTRIC RATES FOR SOUTH ATLANTIC REGION: $.033/KWH OR $9.67/MILLION BTUS.

PARASITIC POWER REQUIREMENTS
- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS.
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS
- AIR HANDLER BLOWER (1/3 HP) 459 WATTS.
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-4. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Atlanta (Nashville 1955 Weather)

	SOLAR PERFORMANCE COOLING ONLY				AUXILIARY BOILER (60% EFFICIENCY FOR GAS)		ECONOMIZER COOLING BTUS/YEAR (HOURS)	COOLING LOAD NOT SATISFIED BTUS/YEAR (HOURS)	TOTAL COOLING COSTS $/YEAR	TOTAL ENERGY COSTS $/YEAR
	SPACE COOLING BTUS/YEAR	SOLAR SYSTEM PUMPS KWHS/YEAR	ABSORPTION SYSTEM PUMPS KWHS/YEAR	ELECTRIC COST $/YEAR	AUX. OUTPUT BTUS/YEAR (HOURS)	GAS (ELECTRIC) $/YEAR				
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	18.1×10^6 x 0.72 (COP)	290	369.7	$21.51	5.7×10^6 (291.88)	$18.56 ($54.82)	1.1×10^6 (457)	0.1×10^6 (54)	$40.07 ($76.33)	$200.33 ($435.74)
NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	18.2×10^6 x 0.72 (COP)	284	362.6	$21.07	5.4×10^6 (281.78)	$17.56 ($51.86)	1.1×10^6 (462)	0.1×10^6 (54)	$38.63 ($72.33)	$187.27 ($401.19)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	18.2×10^6 x 0.72 (COP)	284	362.5	$21.07	5.5×10^6 (281.77)	$17.69 ($52.25)	1.1×10^6 (468)	0.1×10^6 (54)	$38.76 ($73.32)	$185.94 ($395.47)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK NO AUXILIARY COOLING	13.2×10^6 x 0.75 (COP)	172	198.8	$12.10	-	-	1.1×10^6 (463)	3.4×10^6 (548)	$12.10	$258.20 ($336.40)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170 + (T_{AMB}-78°F)/2	17.8×10^6 x 0.74 (COP)	249	314.8	$18.37	5.5×10^6	$17.72 ($52.35)	1.1×10^6 (469)	2,000. (2)	$36.09 ($70.72)	$182.43 ($391.70)
SELECTED SYSTEM INLET TO HOUSE SOLAR COIL BOOSTED TO 125°F - .8676 T_{AMB}	17.8×10^6 x 0.74 (COP)	249	314.7	$18.37	5.4×10^6	$13.14 ($51.76)	1.1×10^6 (464)	2,000. (2)	$31.51 ($70.13)	$194.31 ($418.75)

COP IS SEASONAL COEFFICIENT OF PERFORMANCE.
FEA 1977 NATURAL GAS RATES FOR SOUTH ATLANTIC REGION: $1.94/MILLION BTUS.
FEA 1977 ELECTRIC RATES FOR SOUTH ATLANTIC REGION: $.033/KWH OR $9.67/MILLION BTUS.

PARASITIC POWER REQUIREMENTS
COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
STORAGE LOOP PUMP (1/6 HP) 219 WATTS
AIR HANDLER BLOWER (1/3 HP) 459 WATTS
ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-5. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Albuquerque (1962 Weather)

	SOLAR PERFORMANCE HEATING AND CHARGING STORAGE				AUXILIARY SPACE HEAT (60% EFFICIENCY FOR GAS)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY FOR GAS)		FURNACE BLOWER OPERATION (HEATING AND COOLING)		TOTAL HEATING COSTS $/YEAR
	SPACE HEATING BTUS/YEAR	DOMESTIC HOT WATER BTUS/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	41.7×10^6	14.0×10^6	1392	$50.94	10.5×10^6	$29.07 ($112.61)	4.1×10^6	$8.57 ($44.30)	812	$29.73	$118.31 ($237.58)
NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	37.8×10^6	14.2×10^6	1332	$48.74	9.0×10^6	$24.79 ($96.03)	3.9×10^6	$8.01 ($41.37)	714	$26.12	$107.66 ($212.26)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	38.5×10^6	14.1×10^6	1359	$49.72	8.3×10^6	$22.84 ($88.47)	4.0×10^6	$8.27 ($42.71)	748	$27.39	$108.22 ($208.29)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK NO AUXILIARY COOLING	38.5×10^6	14.1×10^6	1359	$49.73	8.3×10^6	$22.84 ($88.47)	4.0×10^6	$8.27 ($42.71)	748	$27.38	$108.22 ($208.29)

FEA 1977 NATURAL GAS RATES FOR SOUTHWEST REGION: $1.66/MILLION BTUS
FEA 1977 ELECTRIC RATES FOR SOUTHWEST REGION: $.0366/KWH OR $10.72/MILLION BTUS
ALBUQUERQUE 1962 WEATHER

PARASITIC POWER REQUIREMENTS
- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS
- AIR HANDLER BLOWER (1/3 HP) 459 WATTS
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-6. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, Albuquerque (1962 Weather)

	SOLAR PERFORMANCE COOLING ONLY				AUXILIARY BOILER (60% EFFICIENCY FOR GAS)		ECONOMIZER COOLING BTUS/YEAR (HOURS)	COOLING LOAD NOT SATISFIED BTUS/YEAR (HOURS)	TOTAL COOLING COSTS $/YEAR	TOTAL ENERGY COSTS $/YEAR
	SPACE COOLING BTUS/YEAR	SOLAR SYSTEM PUMPS KWHS/YEAR	ABSORPTION SYSTEM PUMPS KWHS/YEAR	ELECTRIC COST $/YEAR	AUX. OUTPUT BTUS/YEAR (HOURS)	GAS (ELECTRIC) $/YEAR				
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	17.1×10^6 x 0.57 (COP)	133	149	$10.33	10,877 (0.10)	$0.03 ($0.12)	4.1×10^6 (379)	-	$10.36 ($10.45)	$128.67 ($248.03)
NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	17.2×10^6 x 0.57 (COP)	133	150	$10.36	6,606 (0.11)	$0.02 ($0.07)	4.1×10^6 (381)	-	$10.38 ($10.43)	$118.04 ($222.69)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	17.2×10^6 x 0.57 (COP)	133	150	$10.35	6,606 (0.11)	$0.02 ($0.07)	4.1×10^6 (381)	-	$10.37 ($10.42)	$118.59 ($218.71)
NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK NO AUXILIARY COOLING	17.2×10^6 x 0.57 (COP)	133	150	$10.35	-	-	4.1×10^6 (381)	1,300. (1)	$10.35	$118.57 ($218.64)

COP IS SEASONAL COEFFICIENT OF PERFORMANCE

FEA 1977 NATURAL GAS RATES FOR SOUTHWEST REGION: $1.66/MILLION BTUS.

FEA 1977 ELECTRIC RATES FOR SOUTHWEST REGION $.0366/KWH OR $10.72/MILLION BTUS.

PARASITIC POWER REQUIREMENTS

COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
STORAGE LOOP PUMP (1/6 HP) 219 WATTS
AIR HANDLER BLOWER (1/3 HP) 459 WATTS.
ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
COOLING TOWER PUMP (1/3 HP) 250 WATTS.

Table C-7. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, New York (1958 Weather)

	SOLAR PERFORMANCE HEATING AND CHARGING STORAGE				AUXILIARY SPACE HEAT (60% EFFICIENCY FOR GAS)		AUX. DOMESTIC HOT WATER (80% EFFICIENCY FOR GAS)		FURNACE BLOWER OPERATION (HEATING AND COOLING)		TOTAL HEATING COSTS $/YEAR
	SPACE HEATING BTUS/YEAR	DOMESTIC HOT WATER BTUS/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	AUX. OUTPUT BTUS/YEAR	GAS COSTS (ELECTRIC) $/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	27.3×10^6	11.9×10^6	1001	$54.55	43.6×10^6	$219.55 ($696.18)	9.2×10^6	$34.76 ($146.96)	1171	$63.81	$372.67 ($961.50)
NO COOLING NO NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK	27.3×10^6	11.9×10^6	989	$53.92	43.6×10^6	$219.64 ($696.44)	9.2×10^6	$34.73 ($146.83)	1116	$60.84	$369.13 ($958.03)
NO COOLING NIGHT SETBACK ABSOLUTE CONTROL OF STORAGE TANK	26.0×10^6	12.2×10^6	951	$51.83	39.9×10^6	$200.70 ($636.40)	8.8×10^6	$33.30 ($140.80)	1015	$55.31	$341.14 ($884.34)
NO COOLING NIGHT SETBACK DIFFERENTIAL CONTROL OF STORAGE TANK	27.2×10^6	11.9×10^6	997	$54.35	38.7×10^6	$194.75 ($617.51)	9.2×10^6	$34.73 ($146.83)	1087	$59.24	$343.08 ($877.94)

FEA 1977 NATURAL GAS RATES FOR NEW YORK/NEW JERSEY REGION: $3.02/MILLION BTUS.
FEA 1977 ELECTRIC RATES FOR NEW YORK/NEW JERSEY REGION: $.0545/KWH OR $15.97/MILLION BTUS.

PARASITIC POWER REQUIREMENTS
- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS.
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS.
- AIR HANDLER BLOWER (1/3 HP) 459 WATTS.
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-8. Hydronic Solar Collector Heating/Cooling System Single-Family Residence, New York (1958 Weather)

	SOLAR PERFORMANCE COOLING ONLY				AUXILIARY BOILER (60% EFFICIENCY FOR GAS)		ECONOMIZER COOLING BTUS/YEAR (HOURS)	COOLING LOAD NOT SATISFIED BTUS/YEAR (HOURS)	TOTAL COOLING COSTS $/YEAR	TOTAL ENERGY COSTS $/YEAR
	SPACE COOLING BTUS/YEAR	SOLAR SYSTEM PUMPS KWHS/YEAR	ABSORPTION SYSTEM PUMPS KWHS/YEAR	ELECTRIC COST $/YEAR	AUX. OUTPUT BTUS/YEAR (HOURS)	GAS (ELECTRIC) $/YEAR				
BASELINE SYSTEM ABSOLUTE CONTROL OF STORAGE TANK INLET TO ABSORBER BOOSTED TO 170°F	$4.[illegible] \times 10^6$ x 0.67 (COP)	51	37	$4.77	0.3×10^6 (12)	$1.46 ($4.66)	0.5×10^6 (196)		$6.24 ($9.43)	$378.91 ($970.93)

COP IS SEASONAL COEFFICIENT OF PERFORMANCE.
FEA 1977 NATURAL GAS RATES FOR NEW YORK/NEW JERSEY REGION: $3.02/MILLION BTUS.
FEA 1977 ELECTRIC RATES FOR NEW YORK/NEW JERSEY REGION: 5.45¢/KWH OR $15.97/MILLION BTUS.

PARASITIC POWER REQUIREMENTS
- COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS.
- STORAGE LOOP PUMP (1/6 HP) 219 WATTS
- AIR HANDLER BLOWER (1/3 HP) 459 WATTS.
- ABSORPTION UNIT PUMP (1/3 HP) 250 WATTS
- COOLING TOWER PUMP (1/3 HP) 250 WATTS

Table C-9. Performance of Solar System With Charging of Storage Tank Using Off-Peak Electric Energy for North Central, New York/New Jersey, Southwest, and South Atlantic Regions

	SOLAR PERFORMANCE			OFFPEAK CHARGING		AUXILIARY SPACE HEAT			AUXILIARY DOMESTIC HOT WATER		HOUSE BLOWER OPERATION		TOTAL ENERGY COSTS $/YEAR
	ENERGY DELIVERED BTUS/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	BTUS/YEAR	ELECTRIC COSTS $/YEAR	PEAK BTUS/YEAR	OFFPEAK BTUS/YEAR	ELECTRIC COSTS $/YEAR	BTUS/YEAR	ELECTRIC $/YEAR	KWHS/YEAR	ELECTRIC $/YEAR	
OMAHA SELECTED SYSTEM WITHOUT CHARGING	57.5×10^6	1145	$157.09	-	-	12.9×10^6	26.1×10^6	$910.30	7.5×10^6	$87.83	1189	$95.12	$1250.34
OMAHA OFFPEAK CHARGING $T_{MIN} = 192. - 1.8\ T_{AMB}$	51.4×10^6	1176	$161.34	48.6×10^6	$569.58	0.1×10^6	0.2×10^6	$4.34	3.9×10^6	$45.38	927	$63.59	$344.13
NEW YORK SELECTED SYSTEM WITHOUT CHARGING	38.8×10^6	997	$217.44	-	-	13.6×10^6	25.1×10^6	$1664.55	9.2×10^6	$146.83	1087	$118.48	$2147.30
NEW YORK OFFPEAK CHARGING $T_{MIN} = 185. - 1.7\ T_{AMB}$	33.6×10^6	1116	$243.26	45.8×10^6	$730.84	0.6×10^6	0.8×10^6	$52.79	6.3×10^6	$100.60	853	$92.99	$1220.48
ALBUQUERQUE SELECTED SYSTEM WITHOUT CHARGING	70.4×10^6	1359	$198.92	-	-	2.5×10^6	5.7×10^6	$169.67	4.0×10^6	$42.71	748	$54.76	$466.06
ALBUQUERQUE OFFPEAK CHARGING $T_{MIN} = 183. - 1.7\ T_{AMB}$	63.5×10^6	1267	$185.46	17.5×10^6	$187.47	0	0	$.00	2.0×10^6	$21.96	547	$40.04	$434.93
ATLANTA SELECTED SYSTEM WITHOUT CHARGING	48.3×10^6	931	$121.40	-	-	7.3×10^6	13.9×10^6	$411.22	5.7×10^6	$54.84	1027	$66.94	$654.40
ATLANTA OFFPEAK CHARGING $T_{MIN} = 211. - 2.1\ T_{AMB}$	42.3×10^6	881	$114.94	30.4×10^6	$289.97	0.2×10^6	0.4×10^6	$9.38	3.2×10^6	$30.17	825	$53.80	$498.26

FEA 1977 ELECTRIC RATES FOR NORTH CENTRAL REGION: 3.43¢/KWH OR $10.05/MILLION BTUS - OFFPEAK.
FEA 1977 ELECTRIC RATES FOR NEW YORK/NEW JERSEY REGION: 5.45¢ KWH OR $15.97/MILLION BTUS - OFFPEAK.
FEA 1977 ELECTRIC RATES FOR SOUTHWEST REGION: 3.66¢/KWH OR $10.72/MILLION BTUS - OFFPEAK.
FEA 1977 ELECTRIC RATES FOR SOUTH ATLANTIC REGION: 3.26¢/KWH OR $9.55/MILLION BTUS - OFFPEAK.
PEAK RATES ASSUMED TO BE FOUR TIMES THE OFFPEAK RATES.

PARASITIC POWER REQUIREMENTS
COLLECTOR LOOP PUMP (1/4 HP) 335 WATTS
STORAGE LOOP PUMP (1/6 HP) 219 WATTS
AIR HANDLER BLOWER (1/3 HP) 459 WATTS

Table C-10. Load Bin - Atlanta
Heating Load Not Satisfied

TIME	JAN	FEB	MAR	APR	MAY	JUN	JUL	AUG	SEP	OCT	NOV	DEC	TOTAL
1.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
2.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
3.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
4.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
5.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
6.0	0.12	0.09	0.05	0.	0.	0.	0.	0.	0.	0.01	0.05	0.12	0.44
7.0	0.28	0.20	0.09	0.	0.	0.	0.	0.	0.	0.02	0.10	0.25	0.93
8.0	0.24	0.14	0.05	0.	0.	0.	0.	0.	0.	0.01	0.03	0.18	0.65
9.0	0.14	0.13	0.05	0.	0.	0.	0.	0.	0.	0.	0.02	0.06	0.40
10.0	0.11	0.12	0.03	0.	0.	0.	0.	0.	0.	0.	0.02	0.02	0.30
11.0	0.05	0.11	0.02	0.	0.	0.	0.	0.	0.	0.	0.02	0.02	0.22
12.0	0.06	0.08	0.00	0.	0.	0.	0.	0.	0.	0.	0.03	0.01	0.19
13.0	0.07	0.07	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.01	0.15
14.0	0.07	0.07	0.	0.	0.	0.	0.	0.	0.	0.	0.00	0.03	0.17
15.0	0.06	0.07	0.	0.	0.	0.	0.	0.	0.	0.	0.01	0.03	0.17
16.0	0.07	0.08	0.	0.	0.	0.	0.	0.	0.	0.	0.04	0.05	0.23
17.0	0.08	0.08	0.	0.	0.	0.	0.	0.	0.	0.	0.04	0.08	0.28
18.0	0.13	0.09	0.	0.	0.	0.	0.	0.	0.	0.	0.04	0.09	0.35
19.0	0.18	0.09	0.00	0.	0.	0.	0.	0.	0.	0.	0.05	0.09	0.42
20.0	0.20	0.11	0.01	0.	0.	0.	0.	0.	0.	0.	0.08	0.15	0.55
21.0	0.20	0.13	0.01	0.	0.	0.	0.	0.	0.	0.	0.09	0.17	0.60
22.0	0.11	0.08	0.01	0.	0.	0.	0.	0.	0.	0.	0.04	0.09	0.32
23.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.
24.0	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.	0.

Fraction of time at hour of day the heating load is not satisfied - summed over each month.

Atlanta selected system with auxiliary cooling boiler used to boost temperatures for heating.

TRADEOFF I GAS AS BACKUP	SOLAR SYSTEM PERFORMANCE HEATING, CHARGING, COOLING					AUXILIARY BOILER PERFORMANCE AT 60% EFFICIENCY			AUXILIARY DOMESTIC HOT WATER PERFORMANCE AT 80% EFFICIENCY		LOAD NOT SATISFIED		AIR HANDLER BLOWERS OPERATION		TOTAL HEATING & COOLING COSTS $/YEAR
	SPACE HEATING BTU 10^6/YEAR	DOMESTIC HOT WATER BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	AUXILIARY GAS INPUT $/YEAR	AUX. OUTPUT BTU 10^6/YEAR	AUX. INPUT $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM INCLUDING NIGHT AND WEEKEND SETBACK NO BOILER	127.5	101.6	74.6	5171	$171	-	-	-	52.7	$64	128.5 $259. (1)	55.4 $134. (2)	6607	$218	$846
WITH BOILER FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	255.4	101.2	95.4	5879	$194	212.9	47.9	$316	53.3	$64	.80	34.6	8494	$280	$854
WITH GAS BOILER TEMPERATURE RESET ALGORITHM	256.1	103.1	129.8	5765	$190	214.2	158.8	$451	50.9	$62	-	-	8470	$279	$982
T_{SET} = 97.5 -.443 T_{AMB} FOR HEATING T_{SET} = 136.5 +. 57 T_{AMB} FOR COOLING															

(1) BASED ON THE GAS FURNACE WITH EFFICIENCY OF .6

(2) BASED ON THE AIR CONDITIONER WITH COP OF 4.0

Table C-11. Hydronic Solar Collector Heating/Cooling System - Commercial Building - Omaha

TRADEOFF I GAS AS BACKUP	SOLAR SYSTEM PERFORMANCE HEATING, CHARGING, COOLING					AUXILIARY BOILER PERFORMANCE AT 60% EFFICIENCY			AUXILIARY DOMESTIC HOT WATER PERFORMANCE AT 80% EFFICIENCY		LOAD NOT SATISFIED		AIR HANDLER BLOWERS OPERATION		TOTAL HEATING & COOLING COSTS $/YEAR
	SPACE HEATING BTU 10^6/YEAR	DOMESTIC HOT WATER BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	AUXILIARY GAS INPUT $/YEAR	AUX OUTPUT BTU 10^6/YEAR	AUX. INPUT $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM INCLUDING NIGHT AND WEEKEND SETBACK NO BOILER	89.3	133.8	131.8	6090	$207	-	-	-	43.0	$82	40.5 $103. (1)	111.7 $278. (2)	6630	$225	$ 879
WITH BOILER FIXED SETPOINT 90°F FOR HEATING 165°F FOR COOLING	129.7	138.8	179.0	7029	$239	40.5	81.7	$311	38.0	$73	-	64.7	9907	$337	$ 960
WITH GAS BOILER TEMPERATURE RESET ALGORITHM	129.7	140.7	243.5	6936	$236	40.5	235.6	$601	45.0	$69	-	.5	9950	$338	$1236
T_{SET} = 94.3 -.3866 T_{AMB} FOR HEATING T_{SET} = 136.5 +. 57 T_{AMB} FOR COOLING															

(1) BASED ON THE GAS FURNACE WITH EFFICIENCE OF .6

(2) BASED ON THE AIR CONDITIONER WITH COP OF 4.0

Table C-12. Hydronic Solar Collector Heating/Cooling System - Commercial Building - Atlanta

TRADEOFF 1. GAS AS BACKUP	SOLAR SYSTEM PERFORMANCE HEATING, CHARGING, COOLING				AUXILIARY BOILER PERFORMANCE AT 60% EFFICIENCY				AUXILIARY DOMESTIC HOT WATER PERFORMANCE AT 80% EFFICIENCY		LOAD NOT SATISFIED		AIR HANDLER BLOWERS OPERATION		TOTAL HEATING & COOLING COSTS $/YEAR
	SPACE HEATING BTU 10^6/YEAR	DOMESTIC HOT WATER BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	AUXILIARY GAS INPUT $/YEAR	AUX. OUTPUT BTU 10^6/YEAR	AUX. INPUT $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM INCLUDING NIGHT AND WEEKEND SETBACK NO BOILER	150.1	170.1	153.9	6486.	$214.	---	---	---	21.3	$26.	16.3 $34. (1)	15.8 $38 (2)	6519.	$215.	$527.
WITH BOILER FIXED SETPOINT 90°F. FOR HEATING 165°F. FOR COOLING	150.1	169.8	154.7	6596.	$218.	26.7	2.7	$ 37.	21.7	$27.	0.2 --	13.4 --	6844.	$226.	$508.
WITH GAS BOILER TEMPERATURE RESET ALGORITHM	150.1	170.1	126.3	6547.	$216.	27.1	72.4	$123	21.3	$26	-- --	.1 --	6847.	$226.	$591.
$T_{SET} = 97.5 - .443^* T_{AMB}$ FOR HEATING															
$T_{SET} = 136.5 + .57^* T_{AMB}$ FOR COOLING															

(1) BASED ON THE GAS FURNACE WITH EFFICIENCY OF .6

(2) BASED ON THE AIR CONDITIONER WITH COP OF 4.0

Table C-13. Hydronic Solar Collector Heating/Cooling System - Commercial Building - Albuquerque

TRADEOFF 1, GAS AS BACK UP	SOLAR SYSTEM PERFORMANCE HEATING, CHARGING, COOLING					AUXILIARY BOILER PERFORMANCE AT 60% EFFICIENCY			AUXILIARY DOMESTIC HOT WATER PERFORMANCE AT 80% EFFICIENCY		LOAD NOT SATISFIED		AIR HANDLER BLOWERS OPERATION		TOTAL HEATING & COOLING COSTS $/YEAR
	SPACE HEATING BTU 10^6/YEAR	DOMESTIC HOT WATER BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	PUMPING POWER KWHS/YEAR	ELECTRIC COSTS $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	AUXILIARY GAS INPUT $/YEAR	AUX. OUTPUT BTU 10^6/YEAR	AUX. INPUT. $/YEAR	HEATING BTU 10^6/YEAR	COOLING BTU 10^6/YEAR	KWHS/YEAR	$/YEAR	
BASELINE SYSTEM INCLUDING NIGHT AND WEEKEND SETBACK NO BOILER	94.4	148.3	43.6	4673.	$262.	---	---	---	86.9	$214	101.5 $416 (1)	53.31 $219 (2)	5061.	$283.	$1394.
WITH BOILER FIXED SETPOINT 90°F. FOR HEATING 165°F. FOR COOLING	195.9	148.8	76.5	5544.	$310.	169.1	81.3	$616.	86.2	$212	---	20.4	7539.	$422.	$1560.
WITH GAS BOILER TEMPERATURE RESET ALGORITHM	195.9	151.1	97.1	5512	$309	169.2	150.3	$786.	83.4	$205.	---	---	7555.	$423.	$1723.
$T_{SET} = 94.3 - .3866^* T_{AMB}$ FOR HEATING															
$T_{SET} = 138.5 + .53^* T_{AMB}$ FOR COOLING															

(1) BASED ON THE GAS FURNACE WITH EFFICIENCY OF .6

(2) BASED ON THE AIR CONDITIONER WITH COP OF 4.0

Table C-14. Hydronic Solar Collector Heating/Cooling System - Commercial Building - New York

TRADEOFF II. ALL-ELECTRK SYSTEM HEATING SEASON	SOLAR PERFORMANCE ENERGY DELIVERED BTUS 10^6/YEAR	PUMPING POWER KWH/YEAR	ELECTRIC COSTS $/YEAR	OFFPEAK CHARGING BTUS 10^6/YEAR	ELECTRIC COSTS $/YEAR	AUXILIARY SPACE HEAT PEAK BTUS 10^6/YEAR	OFFPEAK BTUS 10^6/YEAR	ELECTRIC COSTS $/YEAR	AUXILIARY DOMESTIC HOT WATER BTUS 10^6/YEAR	ELECTRIC $/YEAR	HOUSE BLOWER OPERATION KWH/YEAR	ELECTRIC $/YEAR	TOTAL ENERGY COSTS $/YEAR
OMAHA SELECTED SYSTEM WITHOUT CHARGING	346.2	5171.	$636.	--	--	31.1	97.40	$2500.	42.2	$1900.	6607.	$546.	$ 5582.
OMAHA OFFPEAK CHARGING $T_{MIN} = 172.7 - 1.54\ T_{AMB}$	318.6	5099.	$616.	171.4	$1931.	--	0.58	$ 7.	27.9	$1257.	5544.	$457.	$ 4268.
NEW YORK SELECTED SYSTEM WITHOUT CHARGING	314.9	4673.	$986.	--	--	24.4	77.12	$3339.	69.5	$5316.	5061.	$709.	$10351.
NEW YORK OFFPEAK CHARGING $T_{MIN} = 161.6 - 1.38\ T_{AMB}$	296.1	4648.	$965.	135.7	$2596.	--	0.14	$ 3.	55.0	$4203.	4384.	$613.	$ 8379.
ALBUQUERQUE SELECTED SYSTEM WITHOUT CHARGING	556.0	6486.	$792.	--	--	2.2	14.02	$ 259.	17.0	$ 767.	6519.	$537.	$ 2355.
ALBUQUERQUE OFFPEAK CHARGING $T_{MIN} = 172.7 - 1.54\ T_{AMB}$	537.6	6376.	$778.	43.2	$ 487.	--	--	--	9.4	$ 424.	6135.	$506.	$ 2195.
ATLANTA SELECTED SYSTEM WITHOUT CHARGING	430.5	6090	$764.	--	--	9.1	31.40	$ 786.	43.0	$1995.	6630.	$563.	$ 4109.
ATLANTA OFFPEAK CHARGING $T_{MIN} = 161.6 - 1.38\ T_{AMB}$	402.6	5911.	$735.	70.1	$ 813.	--	--	--	28.0	$1330.	6430.	$547.	$ 3395.

FEA 1977 ELECTRIC RATES FOR NORTH CENTRAL REGION: 3.30/KWH OR $9.67/MILLION BTUS - OFFPEAK
FEA 1977 ELECTRIC RATES FOR NEW YORK/NEW JERSEY REGION: 5.60/KWH OR $16.41/MILLION BTUS - OFFPEAK
FEA 1977 ELECTRIC RATES FOR SOUTHWEST REGION: 3.40/KWH OR $9.96/MILLION BTUS - OFFPEAK
FEA 1977 ELECTRIC RATES FOR SOUTH ATLANTIC REGION: 3.40/KWH OR $9.96/MILLION BTUS - OFFPEAK
PEAK RATES ASSUMED TO BE FOUR TIMES THE OFFPEAK RATES.

PARASITIC POWER REQUIREMENTS
COLLECTOR LOOP PUMP 1 (1 HP) 750 WATTS
STORAGE LOOP PUMPS 2 (2 HP) 1700 WATTS
DELIVERY LOOP PUMP 3 (3/4 HP) 600 WATTS
AIR HANDLER BLOWERS (1/2 HP) 6 X 800 WATTS

Table C-15. Hydronic Solar Collector Heating/Cooling System - Commercial Building